AUTOMOBILE ENGINEER

DESIGN PRODUCTION · MATERIALS

Vol. 47 No. 11 NOVEMBER 1957 PRICE: 3s. 6d.

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to the same standards that have made them America's outstanding line

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FULL STOP

to twenty five double decker buses!

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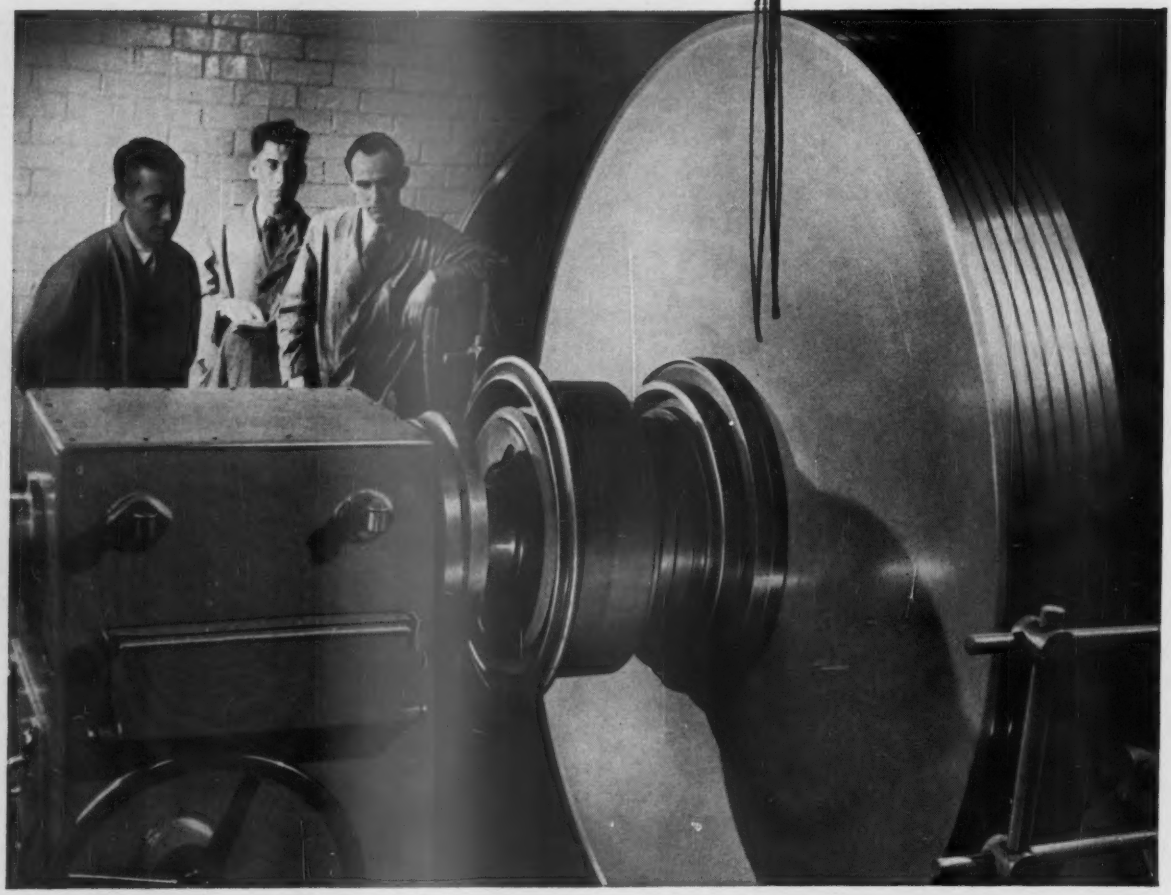


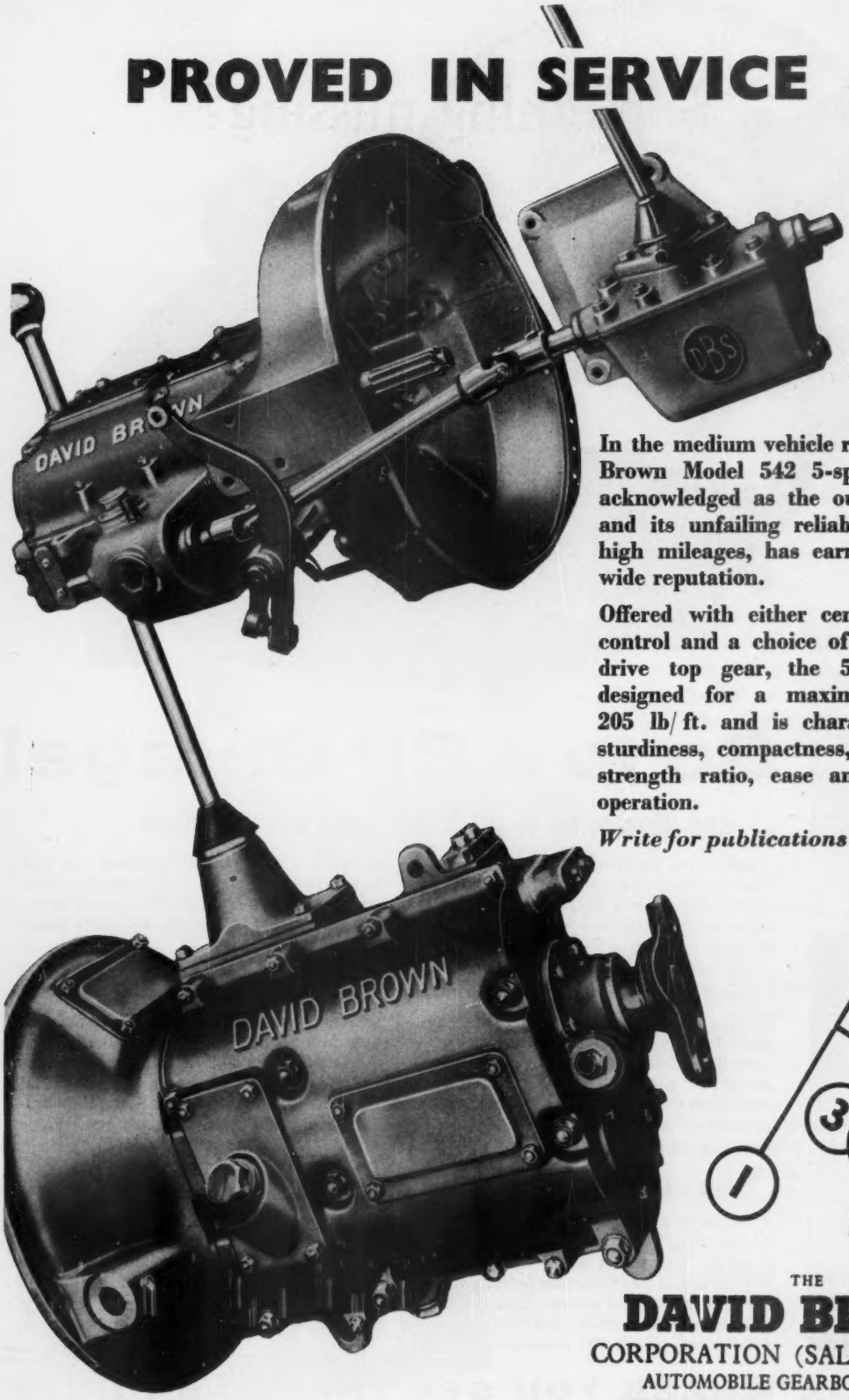
This Mk V Inertia Machine tests Mintex Brake liners under controlled conditions, for friction, fade and general performance. It subjects them to stresses greater than any met on actual service. The flywheels are brought up to a given speed, the brake is applied, and torque, brake drum surface temperature and stopping rate are recorded. The Mk V, one of the largest machines of its kind in the country, generates up to 18½ million ft/lb kinetic energy—equivalent to the energy absorbed in halting 25 double decker buses from a speed of 30 m.p.h. Together with many others similar machines in the B.B.A. laboratories it provides one of the reasons for the long and consistent service that Mintex brake liners give. Research has always been the heart of our business. It continues today with greater emphasis than ever, making sure—and doubly sure—that

you can rely on

MINTEX

Mintex Brake and Clutch Liners are manufactured by British Belting & Asbestos Limited, Cleckheaton, Yorkshire, and are available from MINTEX Service Depots and Distributors throughout the country.

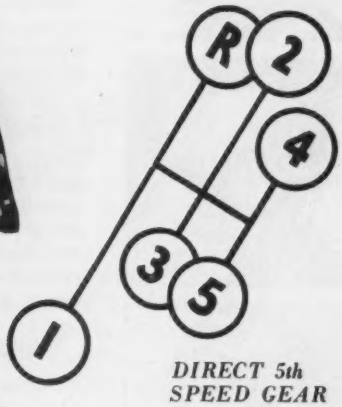
PROVED IN SERVICE



In the medium vehicle range, the David Brown Model 542 5-speed gearbox is acknowledged as the outstanding unit, and its unfailing reliability, over very high mileages, has earned it a world-wide reputation.

Offered with either centre or forward control and a choice of direct or overdrive top gear, the 542 gearbox is designed for a maximum torque of 205 lb/ft. and is characterised by its sturdiness, compactness, low weight to strength ratio, ease and quietness of operation.

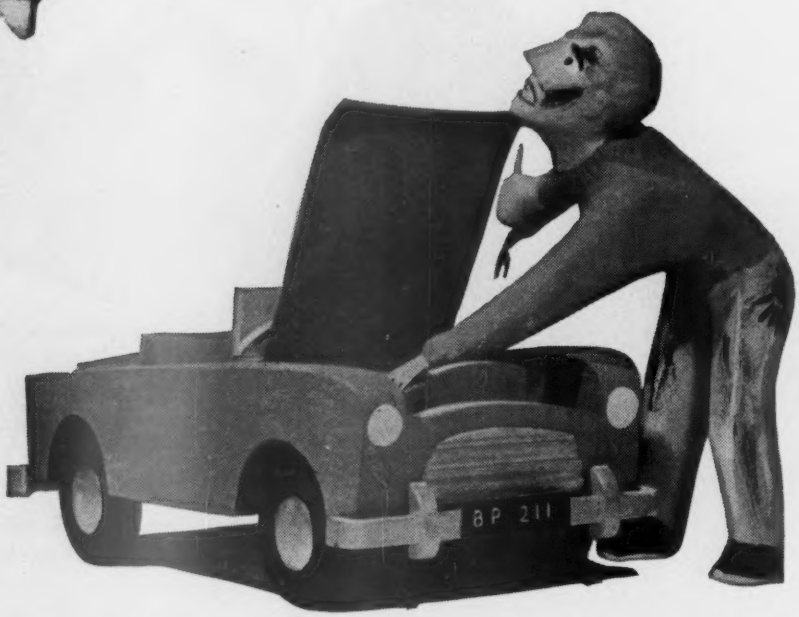
Write for publications E393.26 and 27



DIRECT 5th SPEED GEAR

THE
DAVID BROWN
CORPORATION (SALES) LIMITED
AUTOMOBILE GEARBOX DIVISION
PARK WORKS HUDDERSFIELD

Something missing?



Go to a BP Garage!

. . . and Change Up to BP Super says Stirling Moss

Brilliant racing driver Stirling Moss says "I strongly advise every motorist to go to a BP Garage. You won't find better petrols anywhere. And I particularly want to recommend BP Super–the petrol I've won several major races on. I can definitely promise you that it will give any car a big reserve of extra power, a real zip in acceleration and a better top speed. What's more, it's particularly economical."

. . . and change your oil to BP Energol 'Visco-static' says Ken Gregory

Often Clerk of the Course at race meetings, and Stirling Moss's manager, Ken Gregory, gives this valuable advice. "At a BP Garage you get BP Energol 'Visco-static' motor oil. And this is something I can't recommend too strongly. From personal experience I know this oil gives a tremendous saving in engine wear and petrol consumption, compared with conventional oils. It also helps greatly when starting up from cold. If your car's engine is in good condition, go to your BP Garage and have the oil changed to BP Energol 'Visco-static'. I think it is the most remarkable engine oil on the market to-day."

. . . and enjoy really first class service says Mr. A. J. Meyer

Owner of a 1957 Ford Zodiac, Mr. A. J. Meyer, of 82, Kensington Gardens Square, London, W.2, writes this letter ". . . I am writing to you because I have recently started going to a different garage. This new man sells your BP products:—A BP Solus Station I believe it's called. The treatment I now get is so very much better than I've ever had before that I'm recommending him to all my friends! Service is first class. Fuels and oils are excellent. If publishing this letter will persuade more motorists to 'Go BP' then you have my full permission, and encouragement, to do so!"

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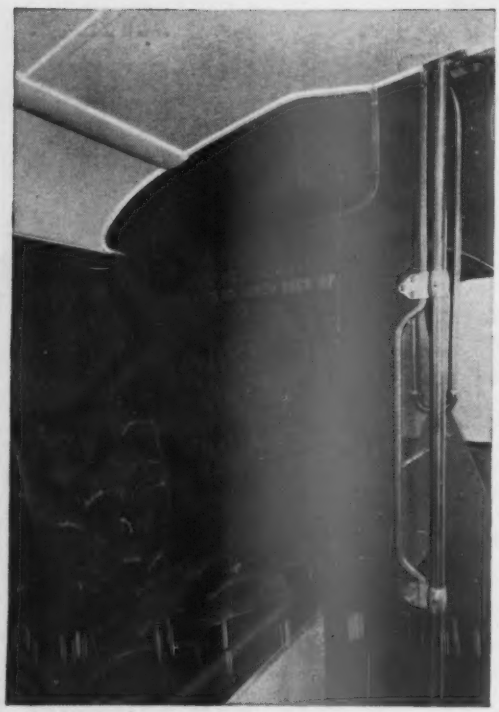
there's room for you inside . . .



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Leathercloth made with Geon PVC is an indefatigable traveller, designed to survive *handsomely* the hard knocks of the travelling public.
It provides the toughest, most durable and yet the most elegant interior trimming for buses, coaches and trains . . .
never loses its excellent surface finish or colour . . .
does not scratch or tear . . .
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'Geon' is a registered trade mark

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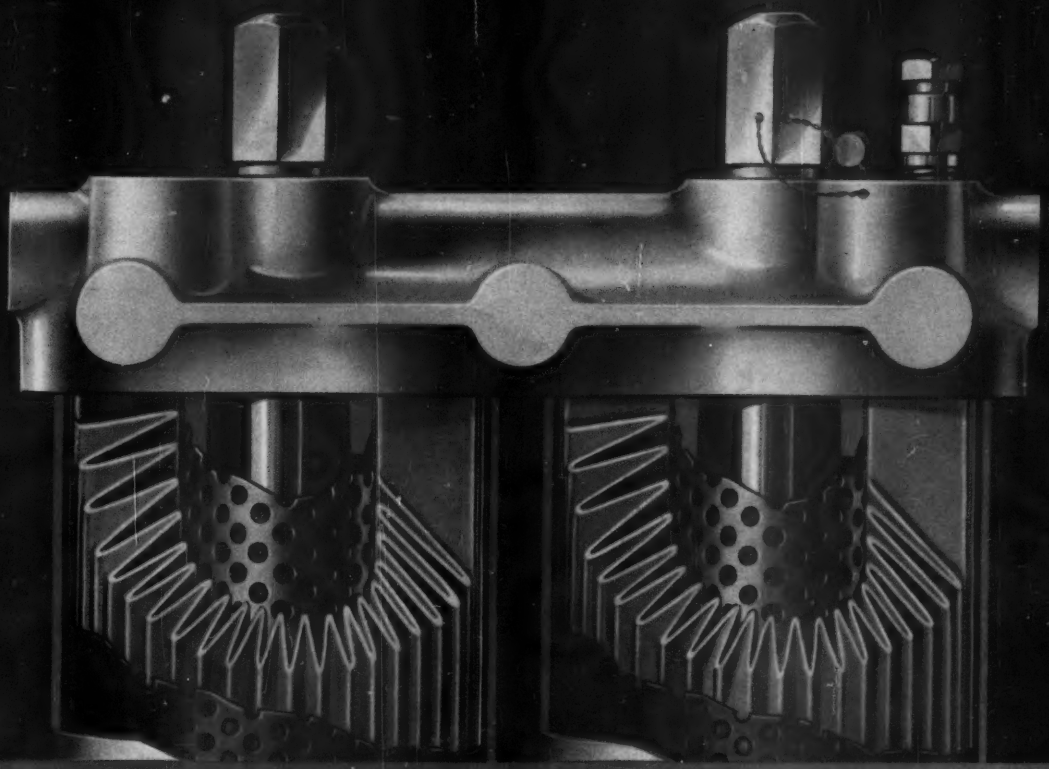

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THIS 'MICRONIC' DIESEL FUEL FILTER

has two stages, each with a 'Micronic' element. The second stage is sealed to discourage interference except at major overhauls, thus constituting a safeguard against the introduction of foreign matter into the injection system when replacing the element of the first stage. This arrangement is recommended where labour is unskilled or the conditions are extremely dusty.

'Micronic' paper filter elements are the result of successful pioneering and development by Purolator.

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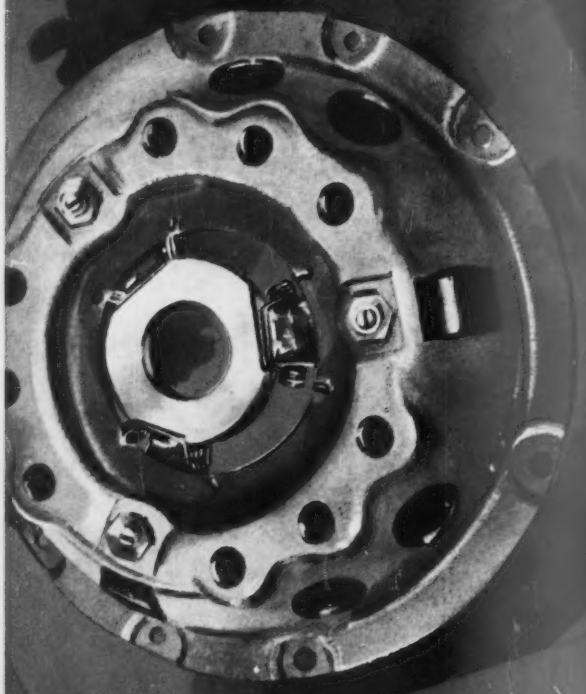
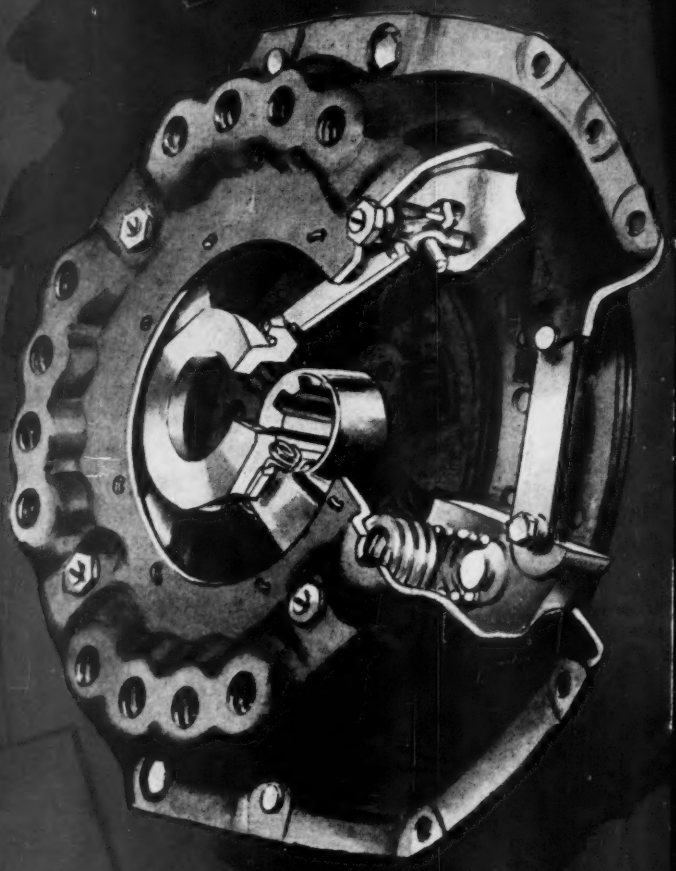
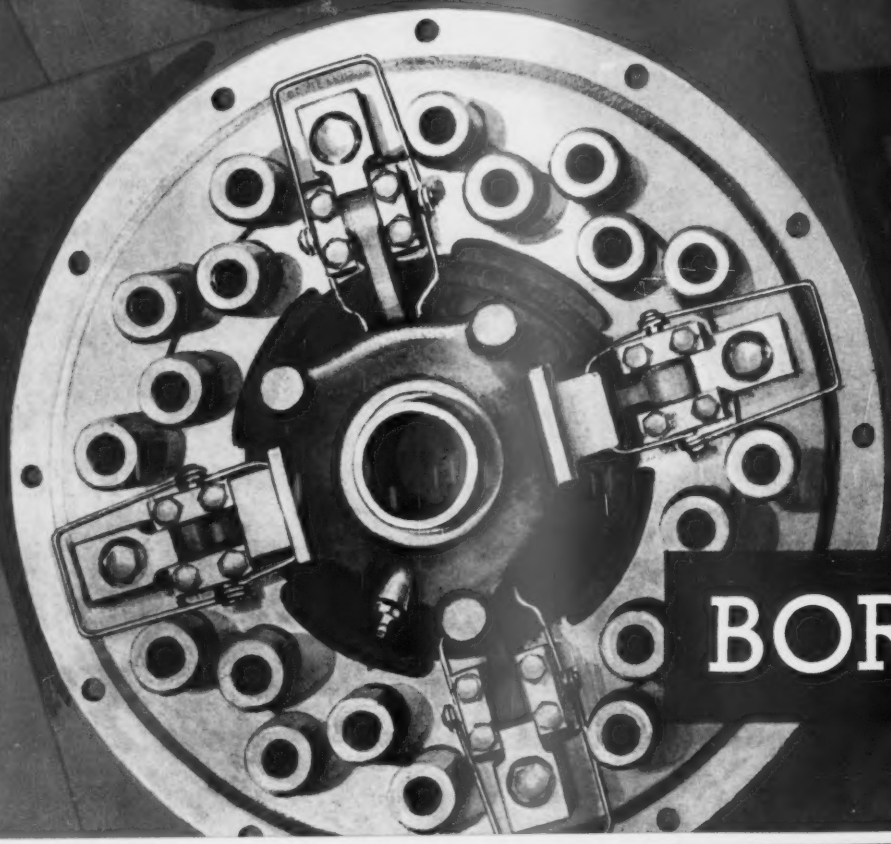
'MICRONIC' LUBRICATING-OIL, FUEL & AIR FILTERS

FRICTION CLUTCHES FOR







Borg & Beck clutches are made in an extensive range, as follows:

'A' TYPE,
6½", 7½", 8", 9", 10" and 11"

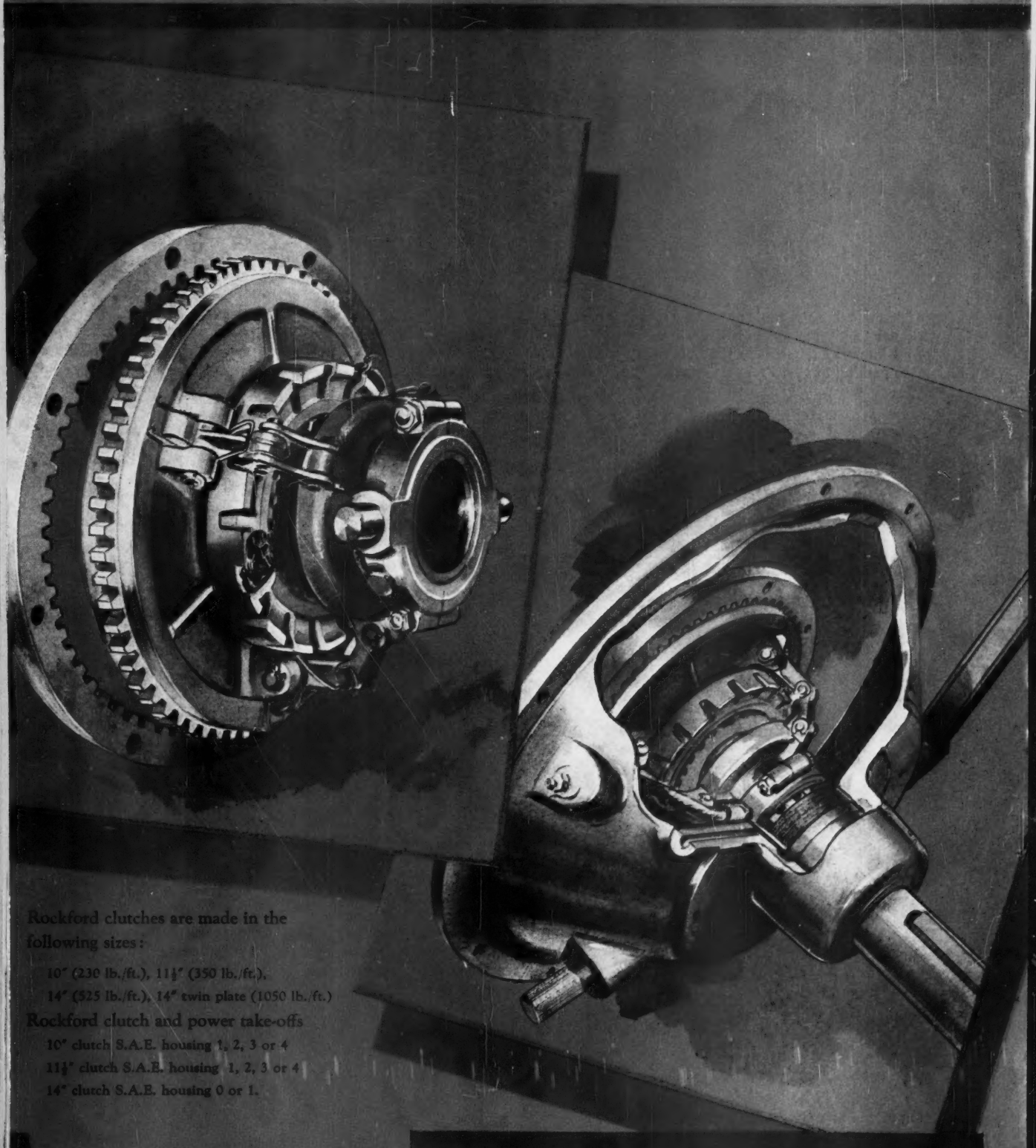
STRAP DRIVE TYPE A.S.,
12" and 13".

'B' TYPE,
14", 16" and 18", single and twin plate.

The torque capacity of these clutches ranges from 38 lb./ft. to 1600 lb./ft.

BORG & BEC

EVERY TYPE OF ENGINE



Rockford clutches are made in the following sizes:

10" (230 lb./ft.), 11¼" (350 lb./ft.), 14" (525 lb./ft.), 14" twin plate (1050 lb./ft.)

Rockford clutch and power take-offs

10" clutch S.A.E. housing 1, 2, 3 or 4

11¼" clutch S.A.E. housing 1, 2, 3 or 4

14" clutch S.A.E. housing 0 or 1.

ROCKFORD

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BORG & BECK COMPANY LIMITED, LEAMINGTON SPA, WARWICKSHIRE, ENGLAND

MAKING STEERING LIGHTER

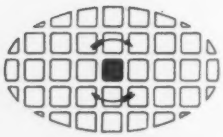
Tyre pressure

The effects of tyre pressure on static and dynamic heaviness are best considered separately.

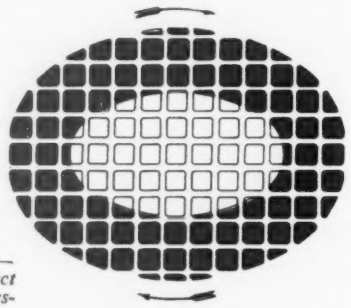
The static torque—parking—we know by experiment to be the highest torque the steering linkage has to contend with. French & Gough ("The Role of Tyres in Automobile Design & Performance", F.I.S.I.T.A. Conference 1954) have quoted the formula:

Torque T: $\frac{\mu}{3}\frac{W^{\frac{3}{4}}}{P^{\frac{1}{4}}}$ i.e. $\frac{\mu}{3}\sqrt{\frac{W^3}{P}}$ *where* μ *is the coefficient of friction between tyre and road,* W *is the load carried and* P *is the inflation pressure.*

We see from this that this static or parking torque is inversely proportional to the square root of the pressure. Therefore (as we all know by experience) the lower the pressure the higher the parking torque. From this point of view therefore the higher the pressure the better. Unfortunately, however, pressure is determined usually by other considerations, such as comfort and handling.



ABOVE—Normal tyre contact—white. Red—contact with gross increase in tyre pressure, giving finger-light steering.



RIGHT—Normal tyre contact—white. Blue—increase in contact with gross reduction in tyre pressure, giving very heavy steering

It is interesting just to take a look at the effect of weight on parking torque, since there has been a very general tendency to increase the proportion of load on the front of the vehicle. If we assume that to please the tyre manufacturers or to keep a constant standard of comfort we put up the pressure in the tyres to maintain the same tyre deflection under load, then, roughly, tyre pressure will vary with load. Applying this relationship to the formula above, we see that parking torque $\propto \frac{W^{\frac{3}{4}}}{W^{\frac{1}{4}}}$, that is W is directly proportional to the load carried by the front tyres.

The effect of tyre pressure on steering heaviness in dynamic conditions depends on the effect of pressure on self aligning torque and on drag. To arrive at the first, on published information (I.Mech.E., A.D. Proceedings, "Tyre Characteristics as Applicable to Vehicle Stability Problems", Hartley & Joy) we have to observe the effect of pressure on self aligning torque at a given slip or drift angle, and of pressure also on the drift angle. For instance, suppose we take an extreme and consider an increase in pressure from 20 to 30 p.s.i. The self aligning torque for a drift angle of 6° would drop from 50 lb. ft. to about 41 lb. ft. But the drift angle to develop a given cornering force would drop from 6° to about 4.8° with the same rise in pressure and this drop in drift angle will drop the self aligning torque further from 41 lb. ft. to 39 lb. ft. An increase in pressure of 50% will therefore result in a drop in self aligning torque, for a given cornering force, of 22%.

When we come to the effect of the drag difference between the two front wheels, an increase in pressure in itself reduces the rolling resistance of a tyre. According to a formula developed by the German "Continental" tyre people, an increase in pressure from 20 p.s.i. to 30 p.s.i. would drop the normal rolling resistance by 28%, i.e. from 49 lb/ton (Hartley & Joy measured figure) to 35 lb/ton. In addition to this reduction itself, there will be a further reduction due to the reduction in drift angle, which will reduce the component of the true sideways force on the tyre which opposes motion of the car. This (Hartley & Joy figures again) would reduce the drag from 167 lb/ton (6° drift) to 130 lb/ton (4.8° drift) so that (as this includes the normal rolling resistance) we would expect the increase in pressure from 20 p.s.i. to 30 p.s.i. to reduce the drag from 167 lb/ton to 116 lb/ton, a drop of about 30%.

It is worth remembering that, since this drag occurs without brakes being applied, it is not acting at the tyre road contact point, but on the centre plane of the wheel at its rotational axis. The offset of this drag force from the king-pin is not therefore the offset at the road but the almost always greater one at the wheel axis height. In spite of this assistance, the drag difference effect is probably not more than a fifth to a quarter of the self aligning torque, and is usually less than this.

It is curious that the percentage reduction is about the same for both static and dynamic tyre torques with increase in tyre pressure.

Thompson
Self-adjusting
STEERING AND SUSPENSION JOINTS
AUTOMOTIVE PRODUCTS COMPANY LIMITED, LEAMINGTON SPA, WARWICKSHIRE, ENGLAND

Three frozen cheers for Electrolux

H'rm . . . H'rm! — said the MD clearing his throat: I have great pleasure in proposing a toast *(CHEERS)* to the Electrolux boys *(LOUD CHEERS)* on the production at Luton of their millionth fridge *(IMMENSE AND PROLONGED APPLAUSE)*. Ladies and gentlemen, only a company of immense skill and enthusiasm in every department could make a million refrigerators for a people who are undoubtedly more cold more often and at more unlikely seasons than any nation in the world. Such a company is the Electrolux Company! *(MORE CHEERS mixed with cries of BRING ON THE TOAST)*. If proof of the far sightedness, the deep wisdom, the high technical intelligence of this Company be asked, I need only say that in every department of Electrolux only the best men, the best materials and THE BEST TOOLS ARE GOOD ENOUGH — AS IS PROVED *(he cried, raising his voice to a fortissimo)* BY THEIR INCREASING USE OF

DESOUTTER TOOLS!

(loud cries of "Oh . . . you self-advertising cad!" and "Push him in the fridge!")

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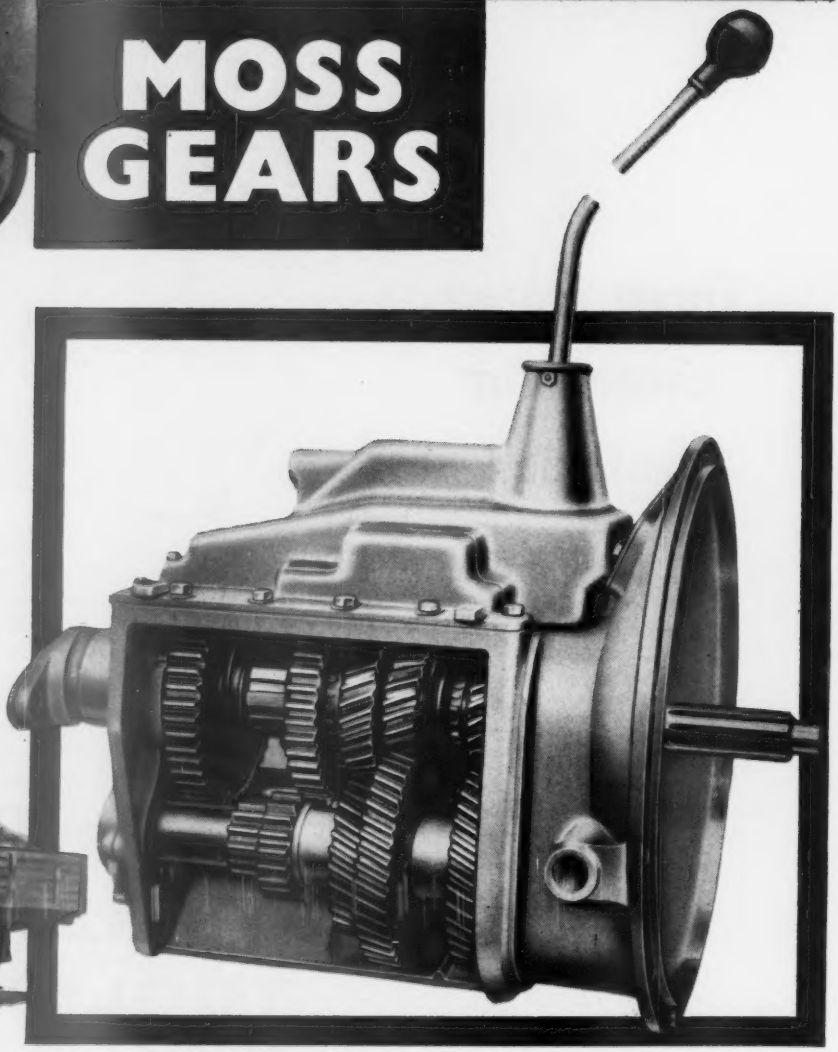
A GREAT 5 speed GEAR BOX!

MOSS GEARS

Suitable for maximum engine torque of 210 lb./ft. at 1,000 r.p.m. Overdrive can be fitted if desired. Full specification and details on application.



PHONE:- ERDINGTON 1661-6 GRAMS:- 'MOSGEAR, BIRMINGHAM'

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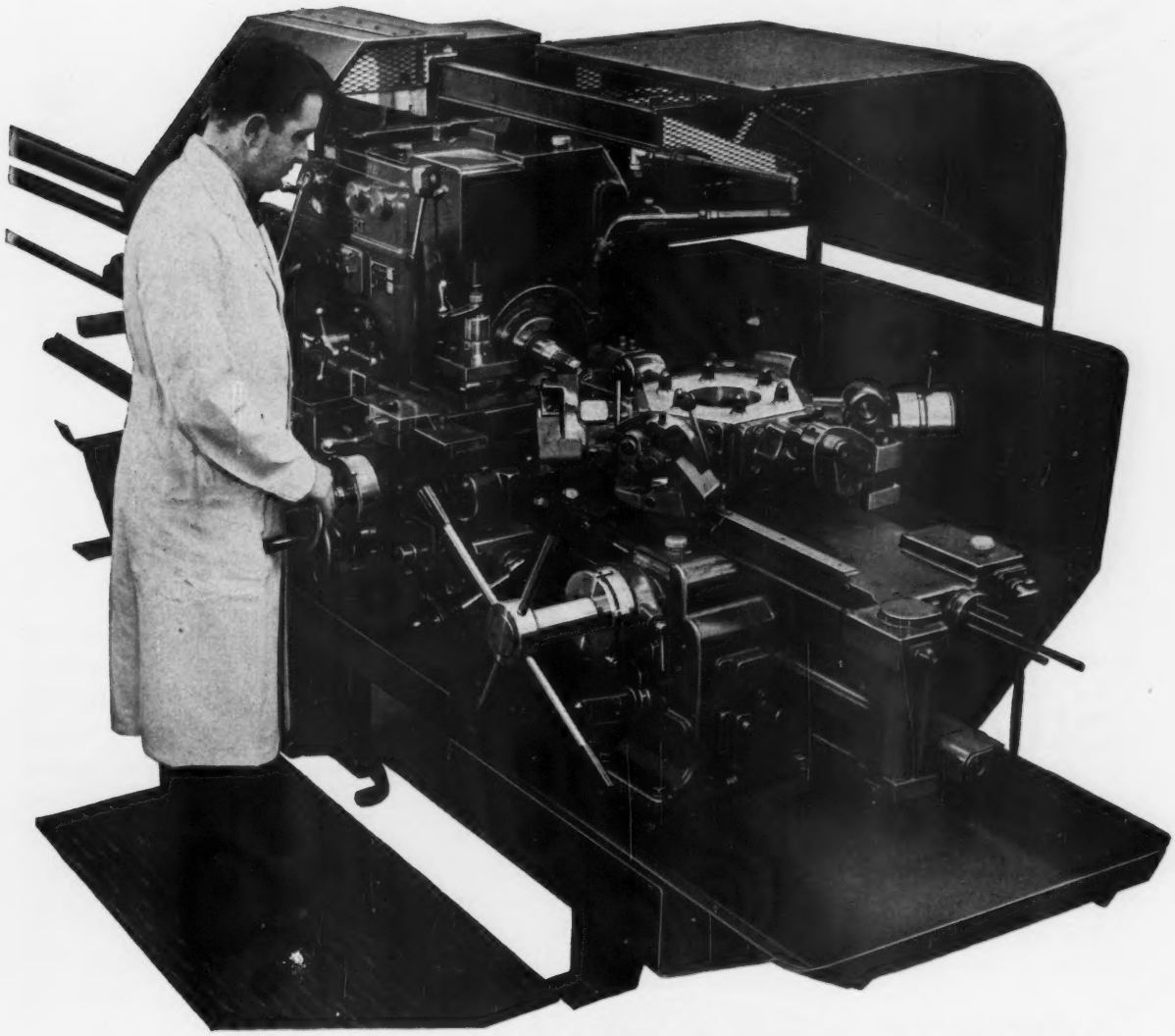
A Masterpiece of Capstan Lathe Design

the HERBERT No. 5

Senior Preoptive



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H.P. of motor	10
16 *speeds* ..	21-1500 *r.p.m.*

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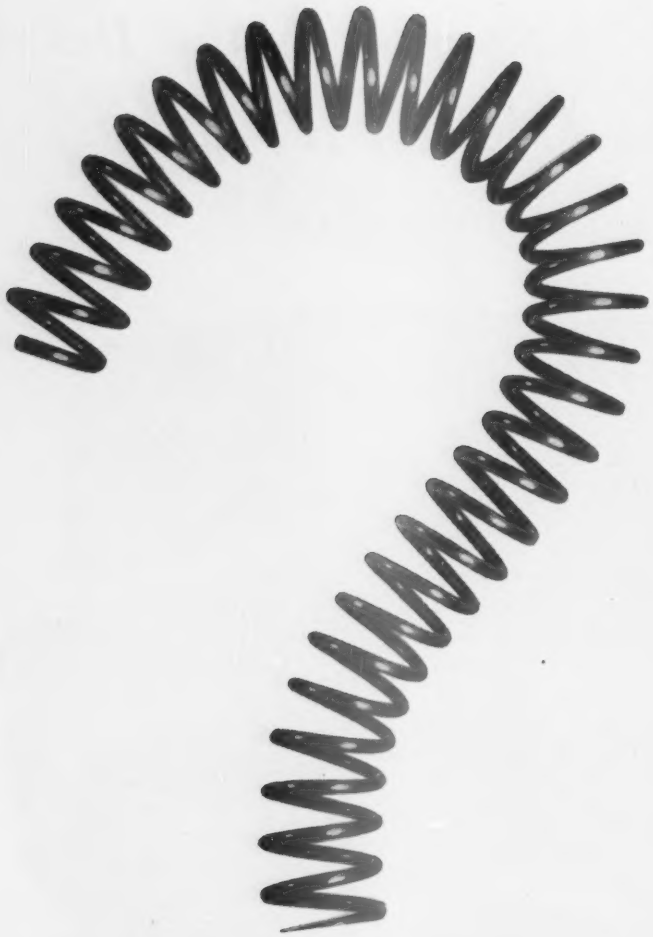

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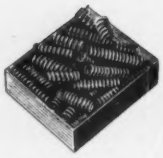
No. 760. 3 doz. Assorted Light Compression Springs 1" to 4" long, 22 to 18 S.W.G., ¼" to ½" diam. 6/6 each.



No. 98A. 3 doz. Assorted 1" to 4" long, ⅛" to ¾" diam., 19G to 15G. 5/6 each.

No. 757. Extra Light Compression, 1 gross Assorted, ¼" to ⅝", ¾" to 2" long, 27 to 20 S.W.G. 15/- each.

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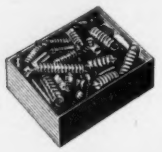

No. 758. Fine Expansion Springs. 1 gross Assorted ⅛" to ⅜", ½" to 2" long, 27 to 20 S.W.G. 15/- each.

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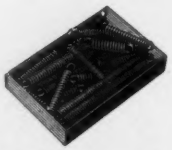

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Cut Production costs with TERRY Wire Circlips ***(Square Section)***

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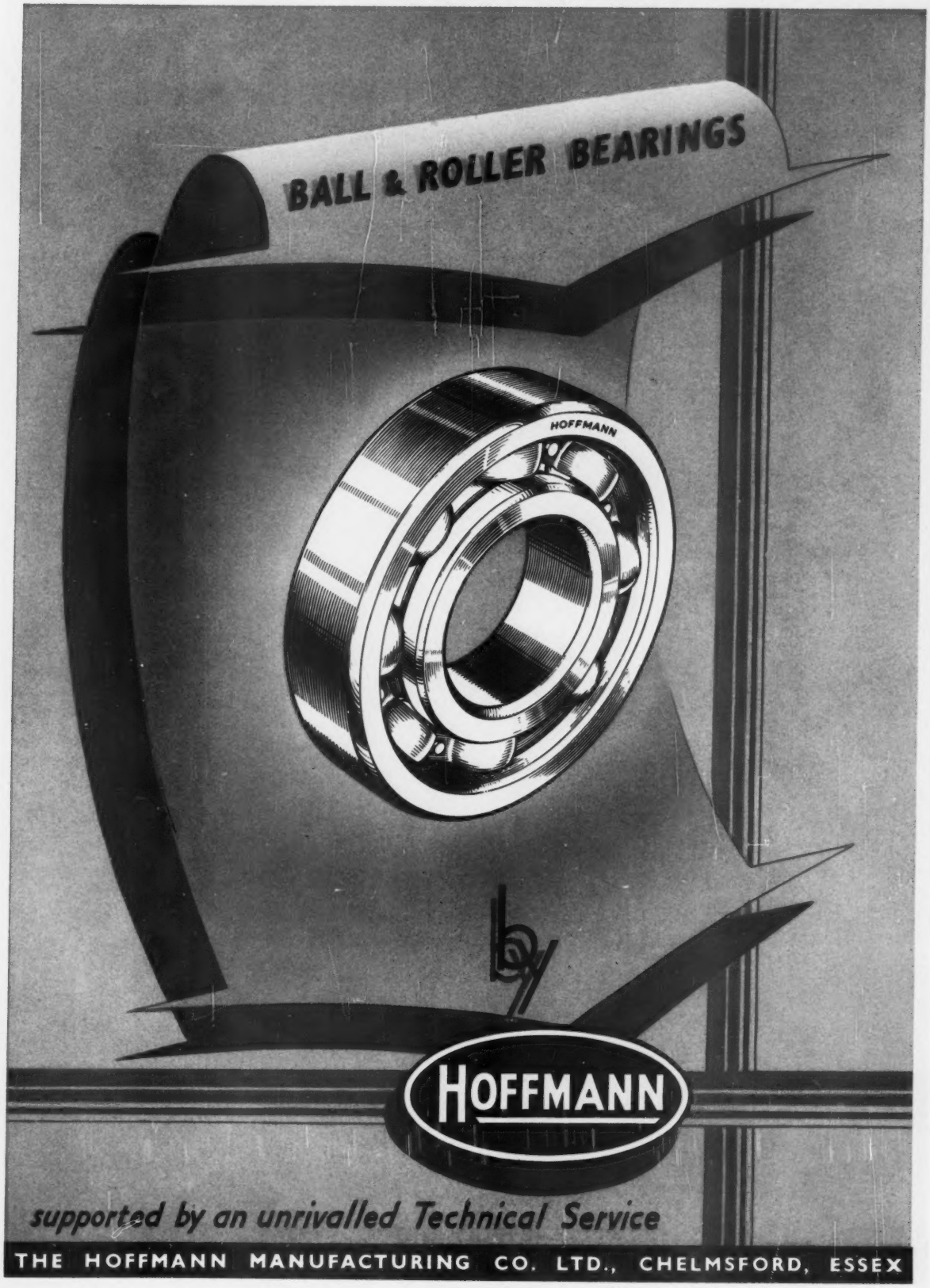
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by
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THE HOFFMANN MANUFACTURING CO. LTD., CHELMSFORD, ESSEX

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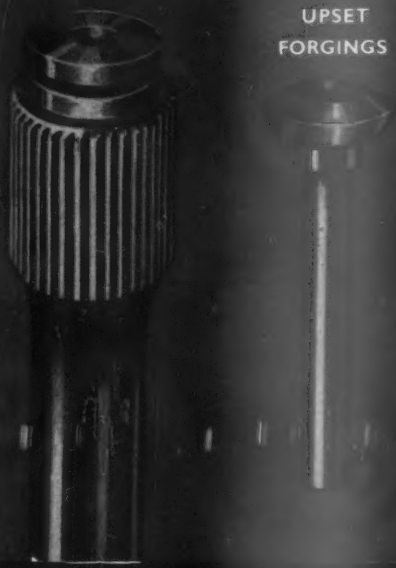
TOLEDO WOODHEAD SPRINGS
give top performance everywhere



TORSION BARS

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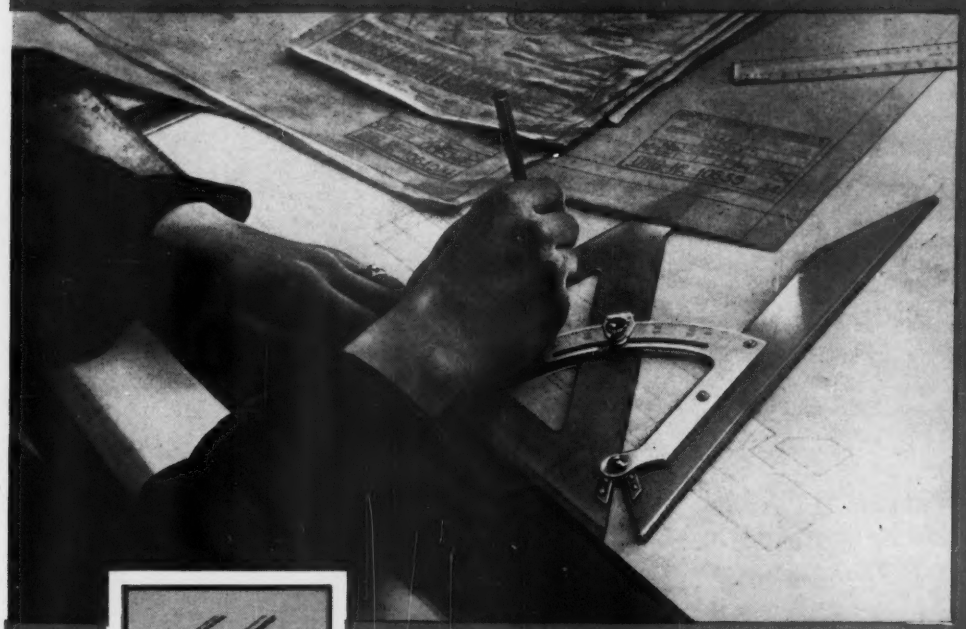
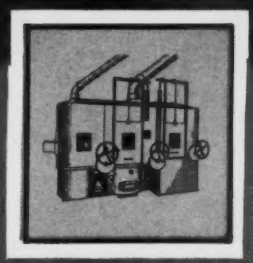
. . . lies behind every recommendation made by the 'Cassel' Heat-treatment Service. A complete range of salts and salt-bath furnaces is manufactured to meet a wide variety of heat-treatment problems.





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The 'Cassel' Process offers these advantages:

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- *Uniform results easily obtained*
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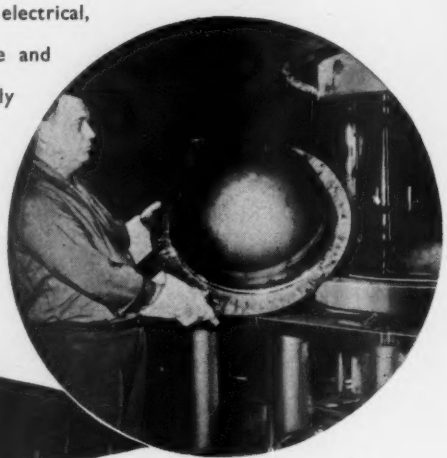
Things forged, extruded, drawn, hammered, spun, milled, broached, drilled; things colossal and things microscopic; things nuclear, plastic, electronic, electrical, ferrous and non-ferrous; things present and futuristic: all these and more are our concern. If any are your concern too, we can probably help, either by solving a lubricating problem or by making a good process even more efficient and economical.



If you would like a general picture of our range, write to our Richmond office for Bulletins 98 (electrical and electronic applications) and 99 (metalworking and forming)

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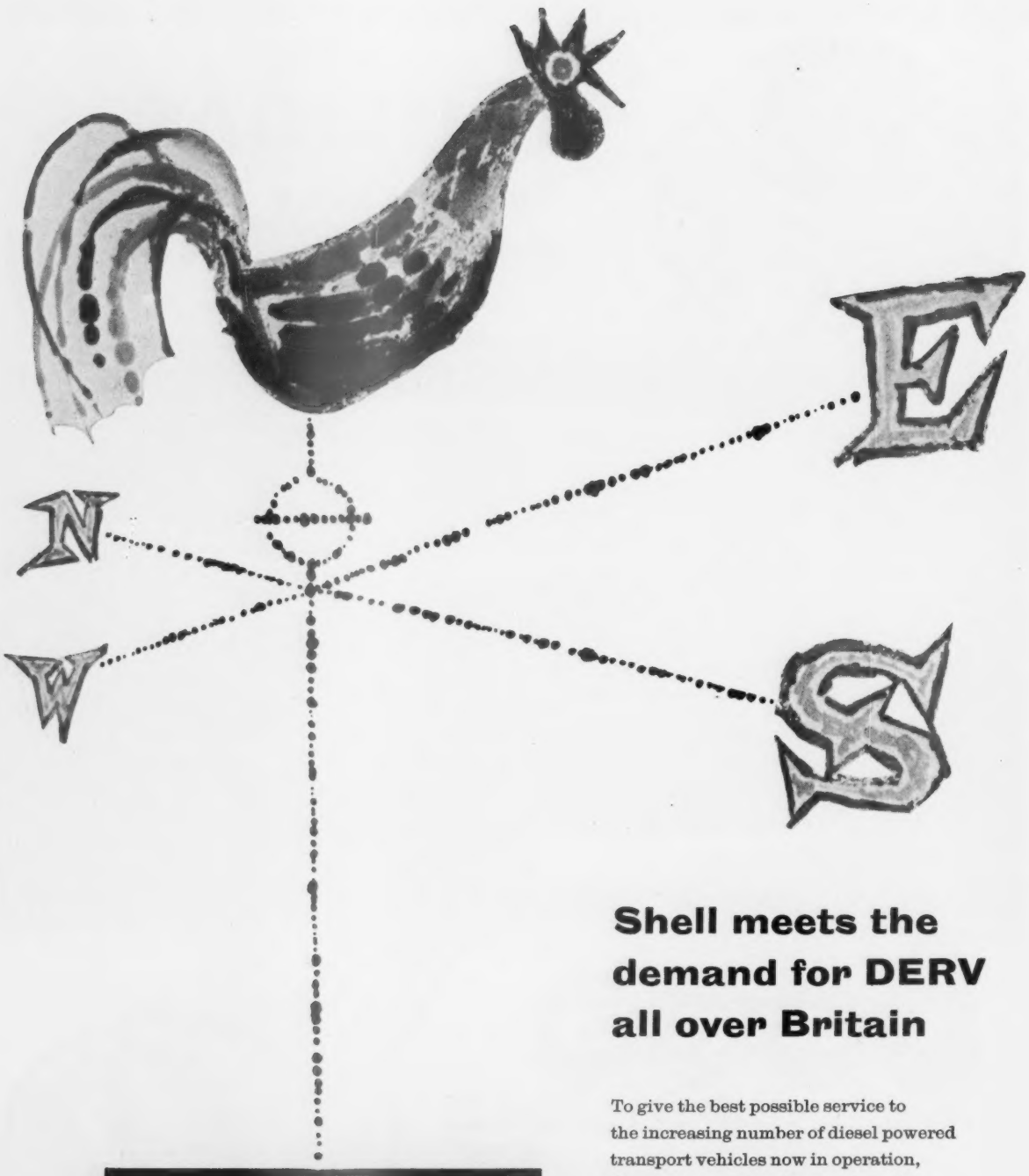

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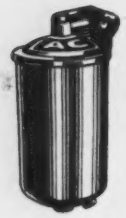
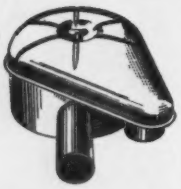
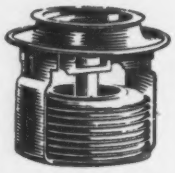


AC OIL FILTERS

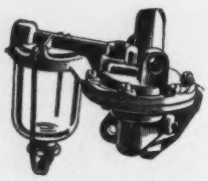


AC AIR CLEANERS AND SILENCERS

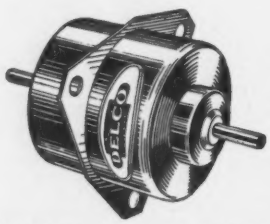


AC THERMOSTATS

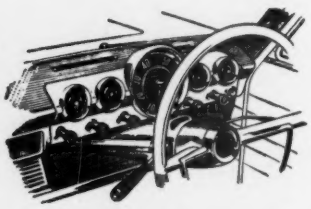
AC SPARK PLUGS



AC FUEL PUMPS



DELCO ELECTRIC MOTORS



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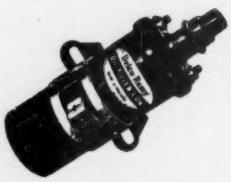
Virtually every British car has one or more AC-Delco components designed into it on the drawing-board. AC-Delco research engineers will gladly help when you are planning a new design or modifying an existing one. Liaison at the drawing-board stage can help your design and production programme.



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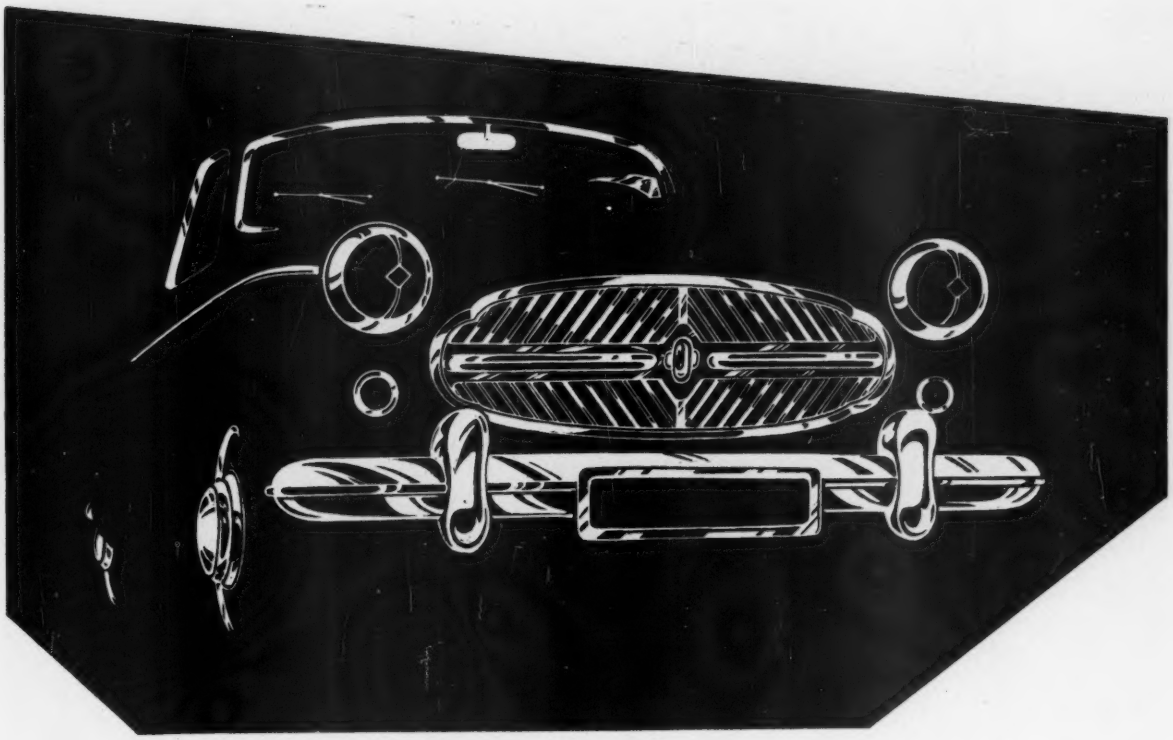

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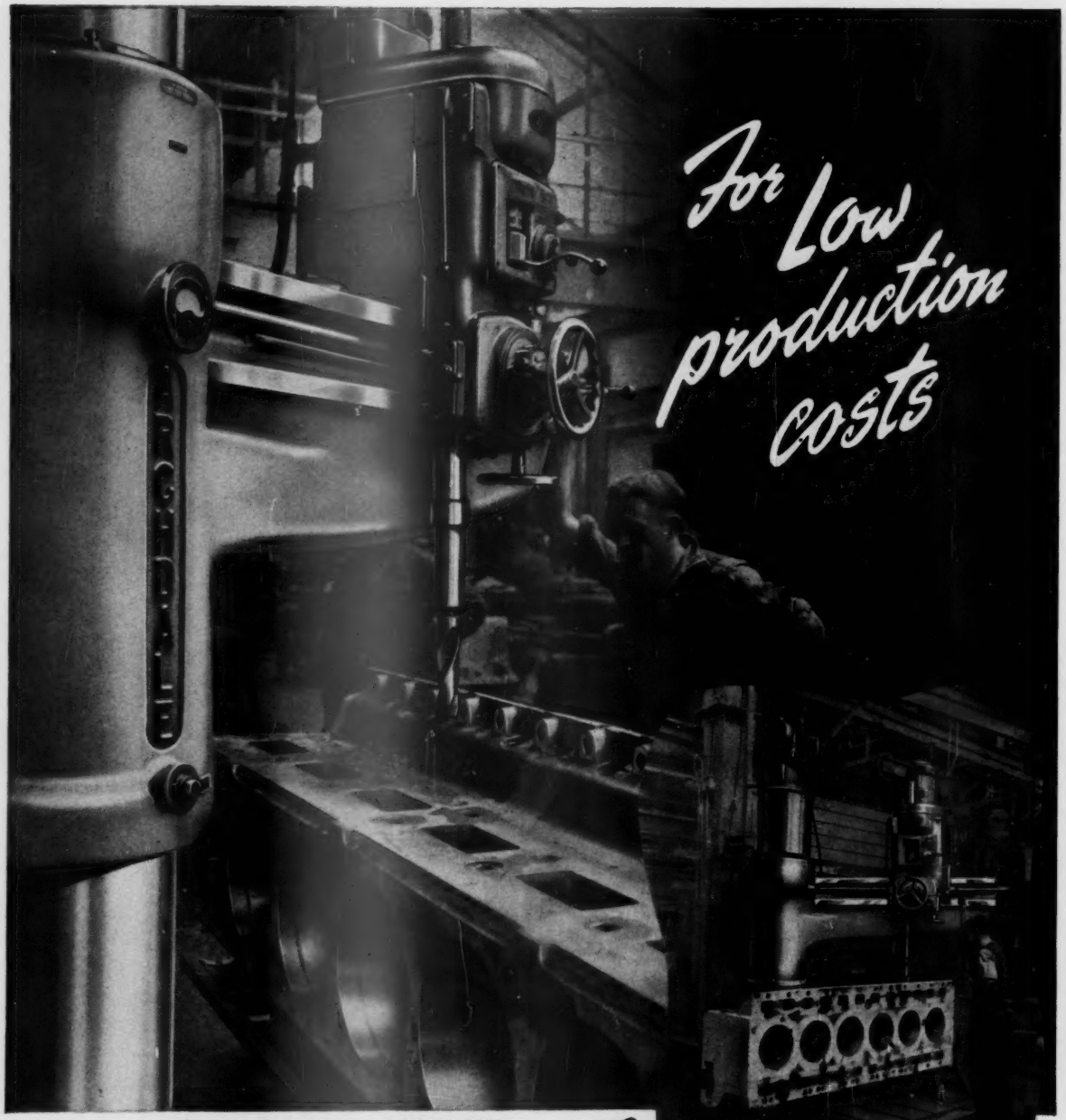

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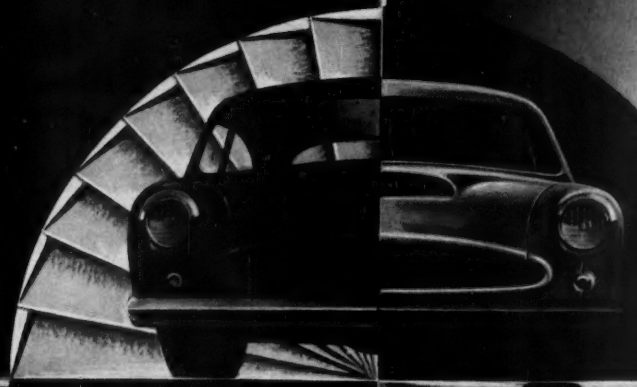

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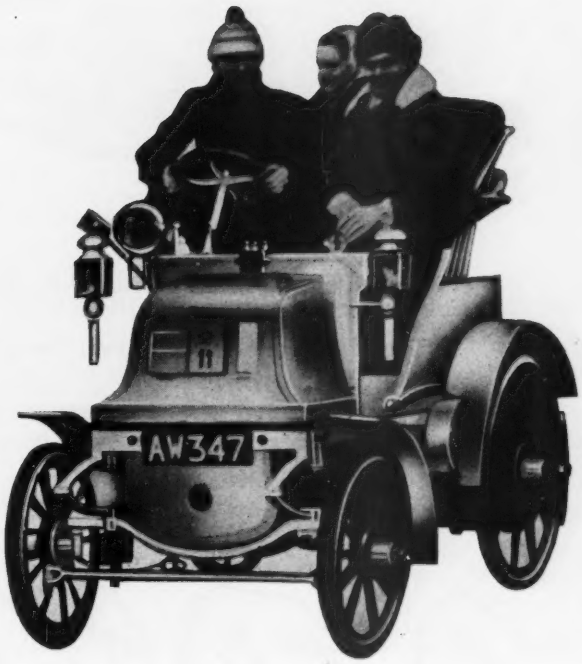

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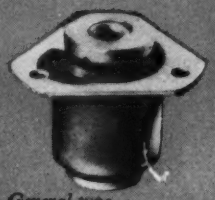
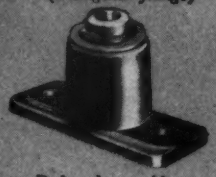
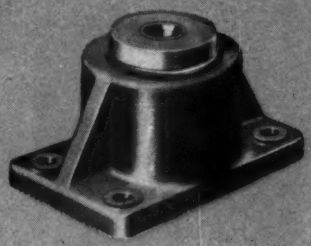


General type (triangular flange)

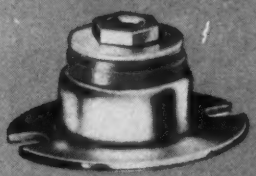


Pedestal assembly

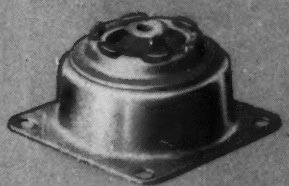


Pedestal mounting

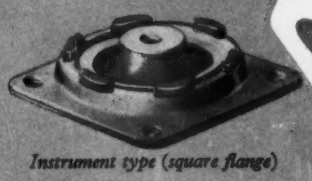
Frustacon type



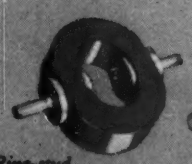
FN type



Instrument type (pedestal flange)



Instrument type (square flange)



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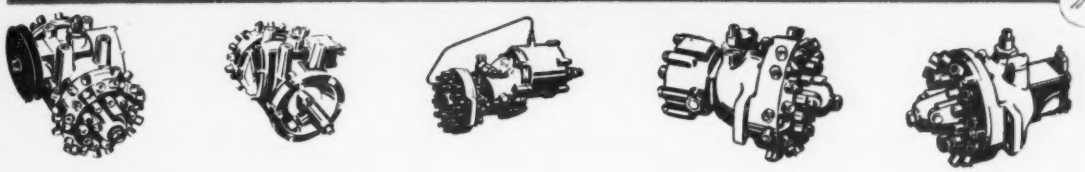

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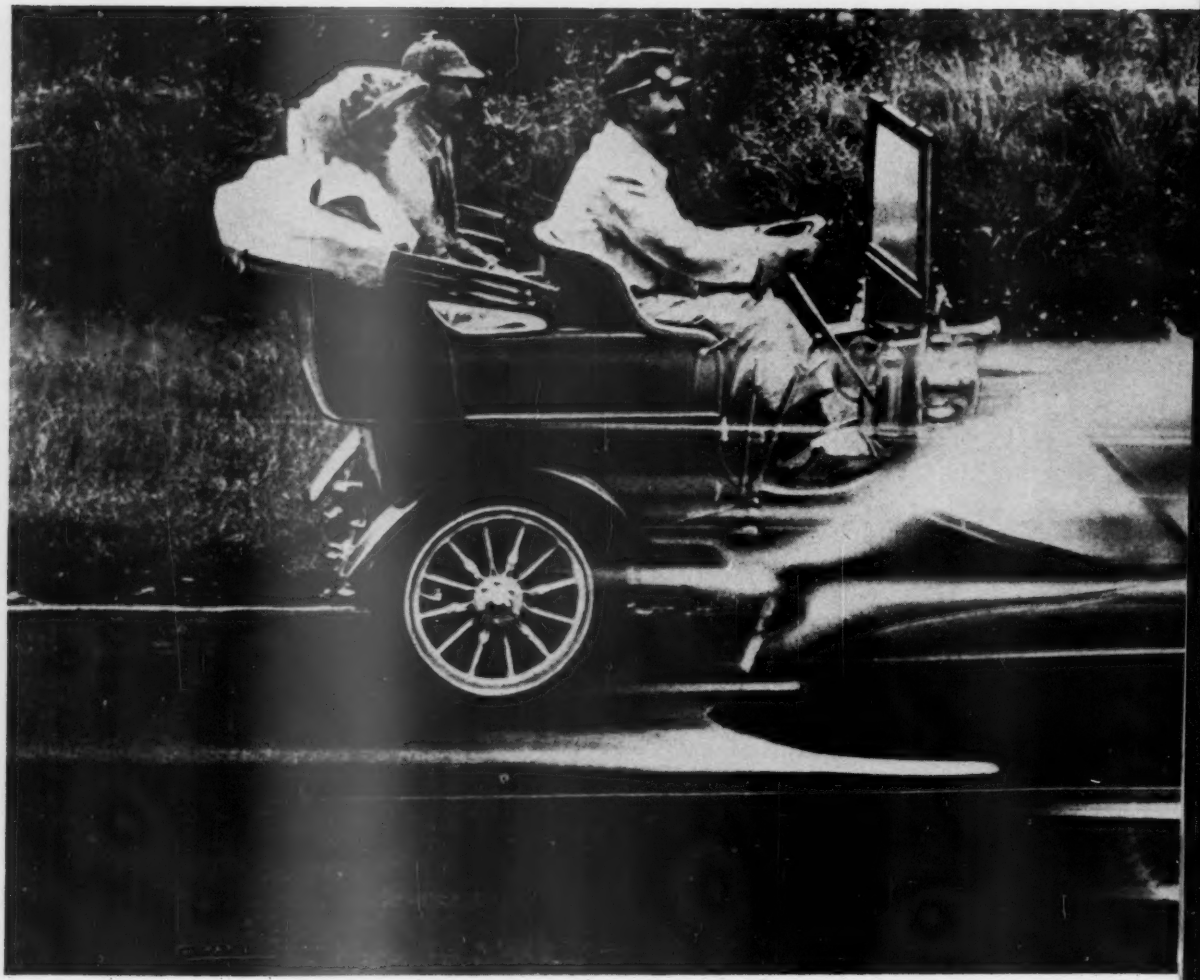

The history of the motor car is the history of **FERODO**

In 1895 the first car had arrived in Britain. In 1896 the three-mile an hour speed limit and the red flag rule had been repealed. Herbert Frood inventor of Ferodo brake linings began his investigations into friction materials in 1897. So the history of the motor car and of Ferodo began together.

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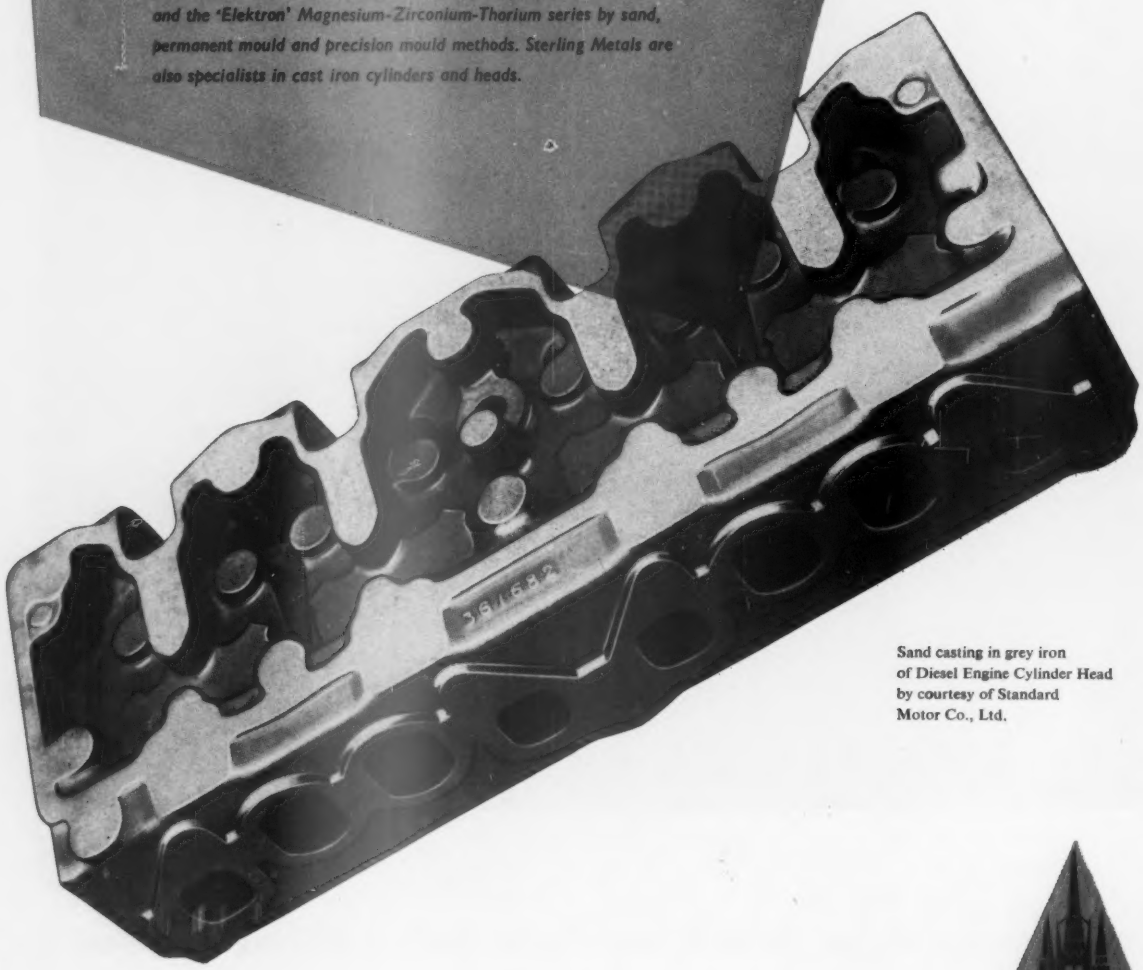

Sand casting in grey iron of Diesel Engine Cylinder Head by courtesy of Standard Motor Co., Ltd.

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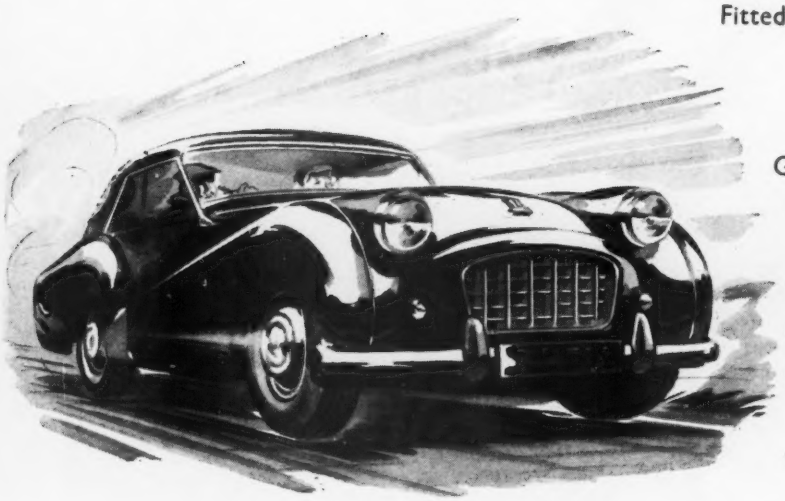

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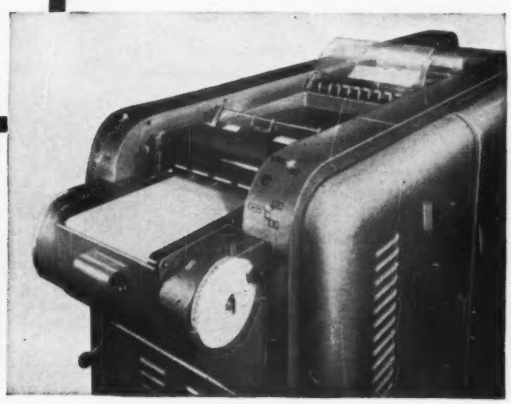

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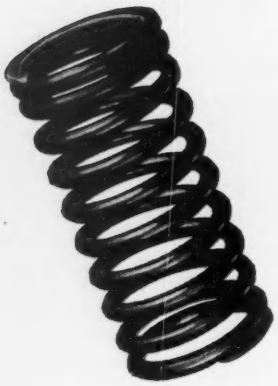

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. . . from your point of view and from ours the focus is on reliability. If a spring passes our tests it will pass any that can be imposed and particularly the test of time.

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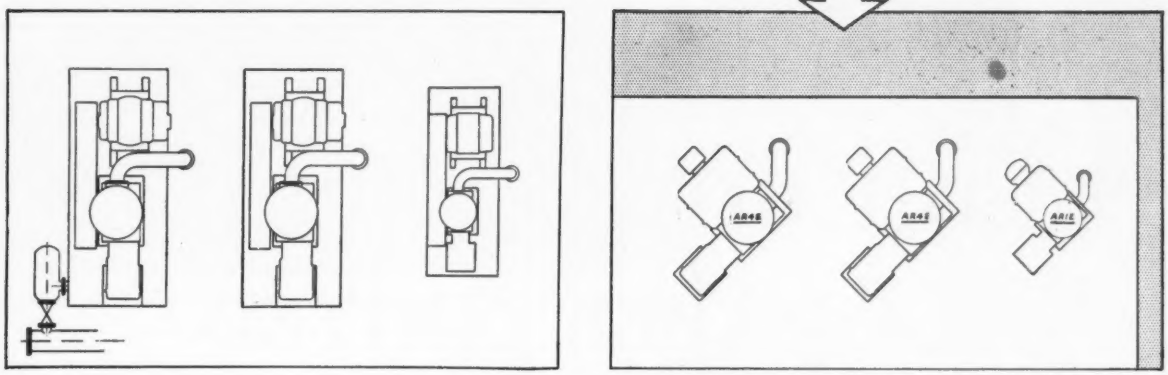


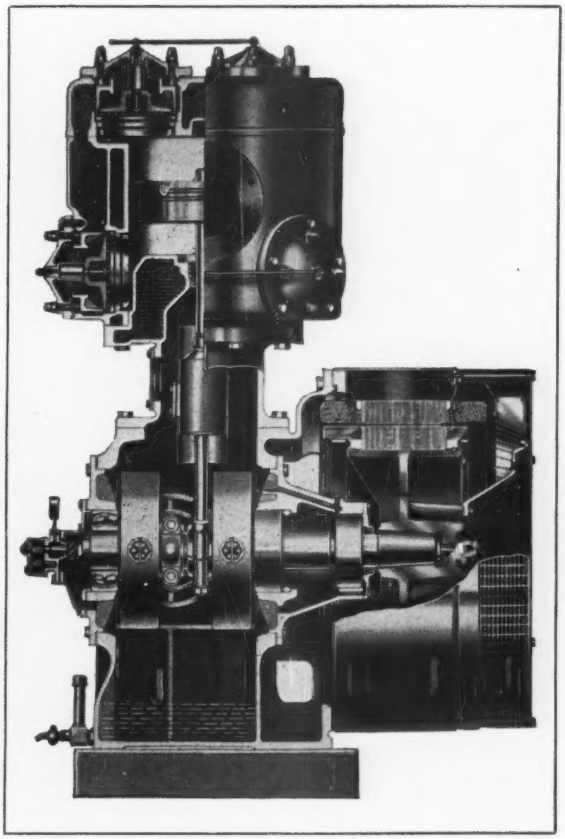
The shaded area indicates the space you save by installing Atlas Copco AR-E 'Power-Pack' compressors in place of standard machines

with Atlas Copco 'Power-Pack' Compressors

The Atlas Copco AR-E 'Power-Pack' range is a series of heavy-duty stationary compressors, specifically designed to cut down on floor space and installation costs. The diagrams above show how these compact machines save on space—up to 25% in this typical installation.

Installation costs are cut because AR-E compressors are delivered complete. The motor is an integral part of the unit, mounted on the compressor casing. Putting an AR-E to work takes virtually no time at all—particularly when mounted on skidframes, when bolting-down is unnecessary and the machines can easily and quickly be moved.

Maintenance is reduced because there is no vee-belt or coupling, and the motor is bolted directly to the crankcase, eliminating risk of misalignment.



World-wide sales and service

AR-E 'Power-Pack' compressors are just a few of the great range of compressed air machines manufactured by the Atlas Copco Group, and sold and serviced throughout the world. For further details, contact your local Atlas Copco Company or Agent, or write to Atlas Copco AB, Stockholm 1, Sweden, or Atlas Copco (Great Britain) Limited, Beresford Avenue, Wembley, Middlesex.

Atlas Copco *Manufacturers of Stationary and Portable Compressors, Rock-Drilling Equipment, Loaders, Pneumatic Tools and Paint-Spraying Equipment.* C.18

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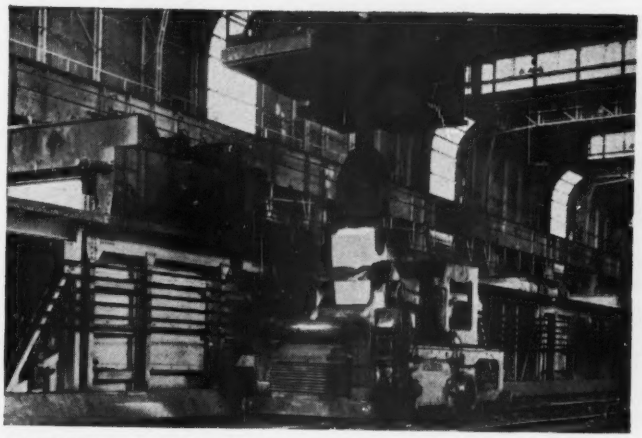
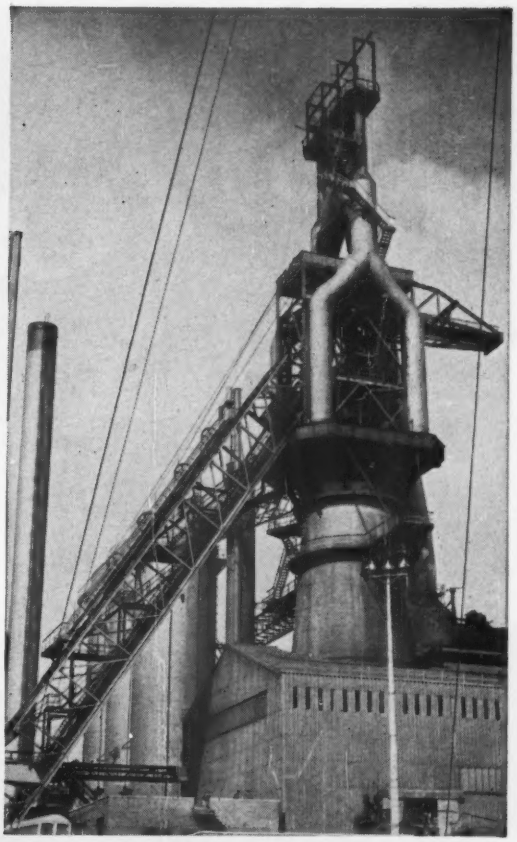
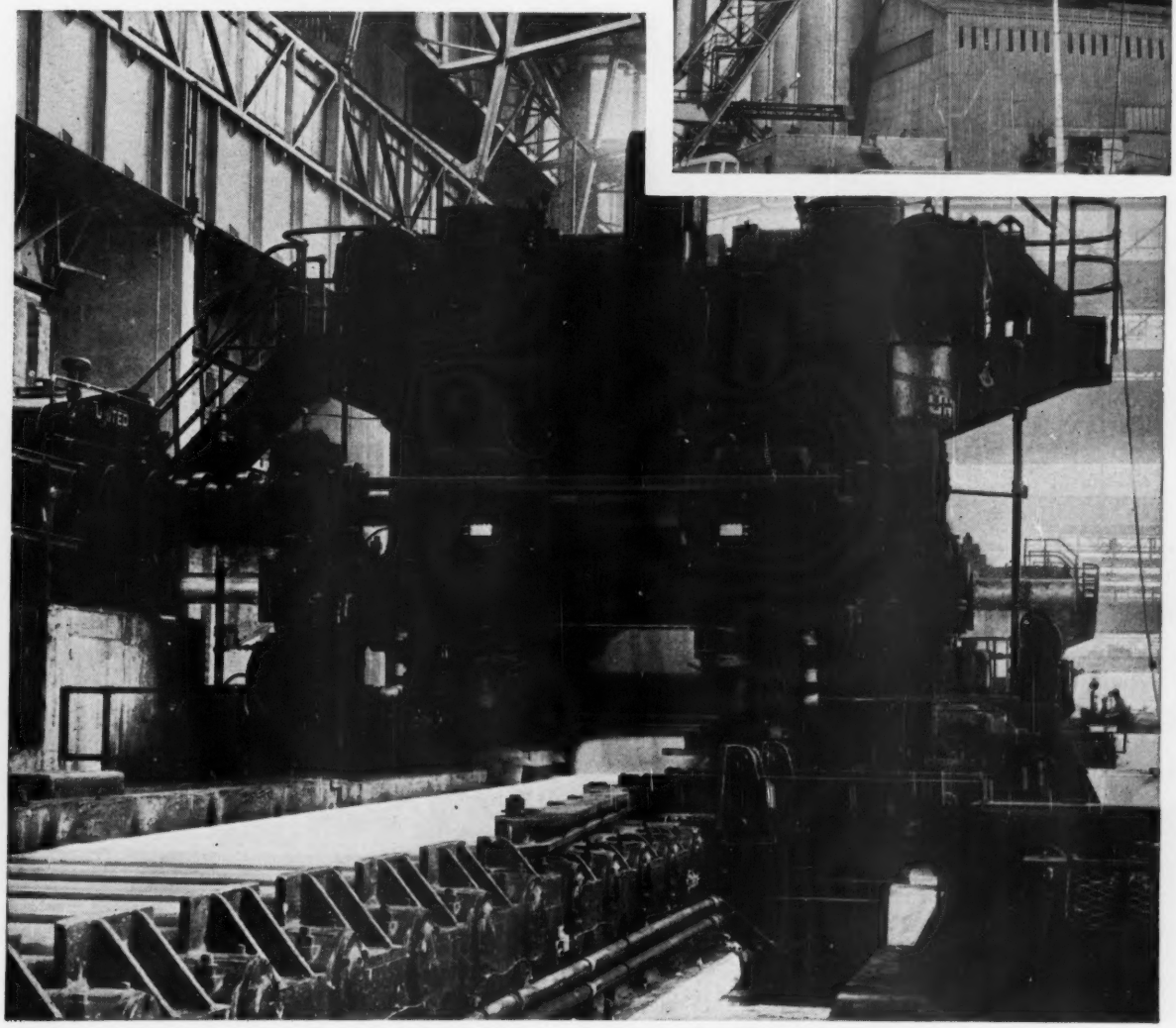


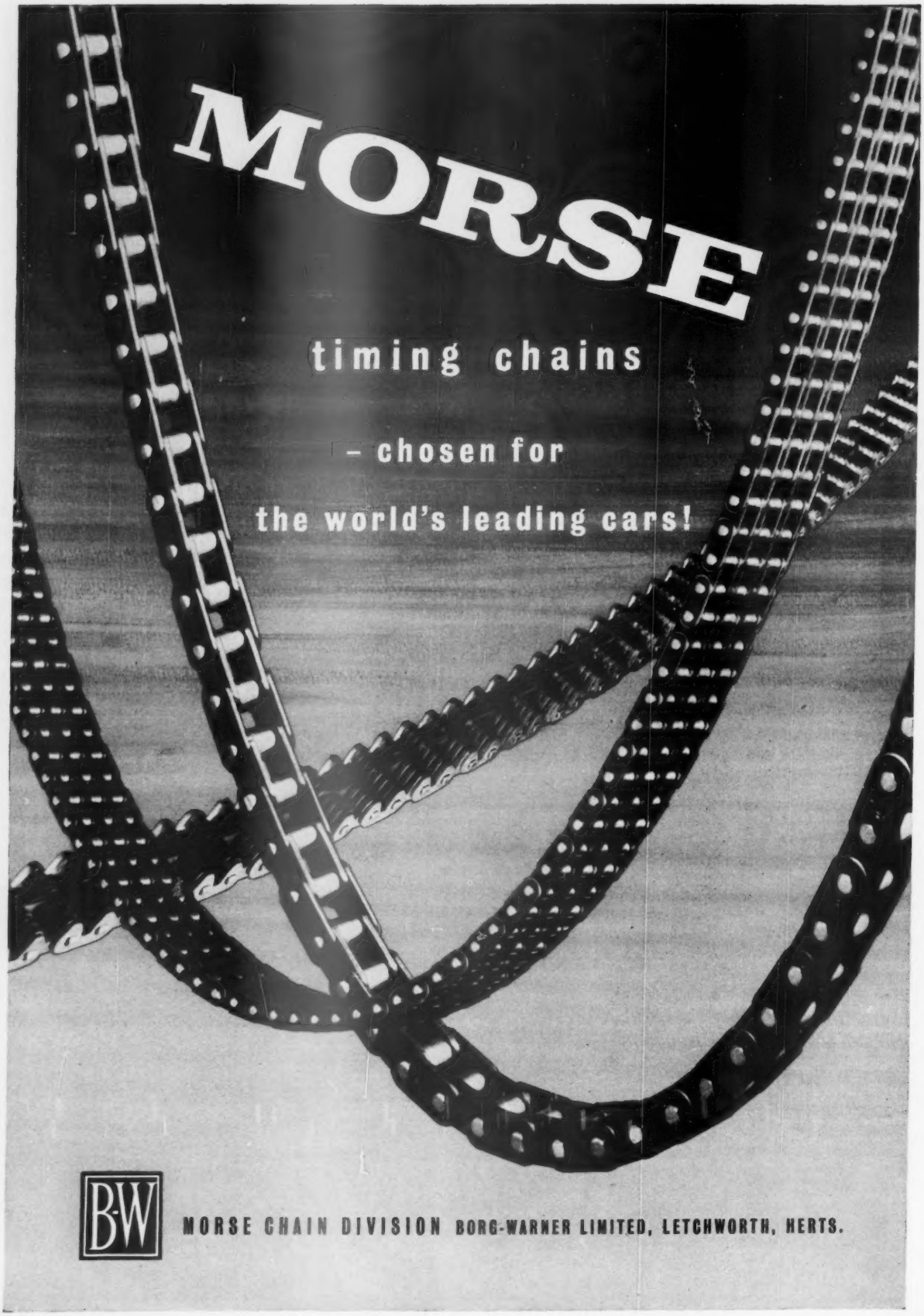
After leaving the soaking pits: an ingot being placed on the ingot buggy



No. 4 Blast Furnace

A slab passing through one of the roughing stands of the 80" continuous hot-strip mill at Abbey Works

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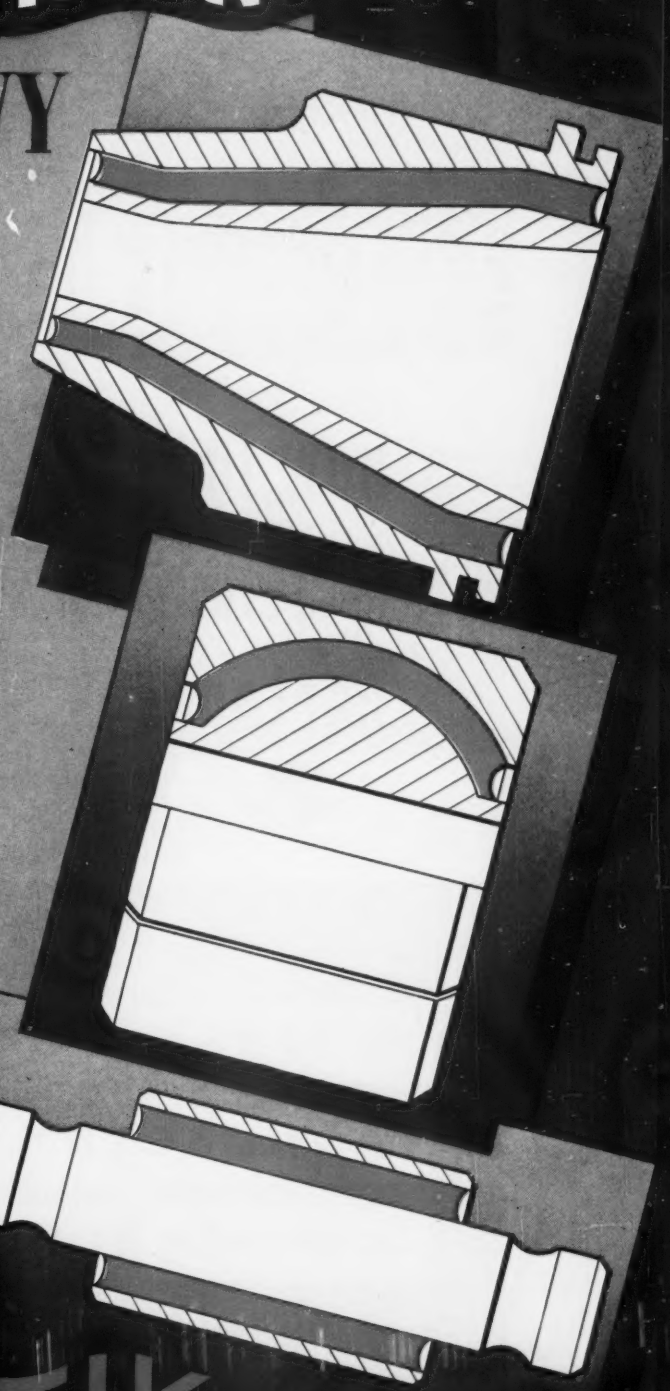

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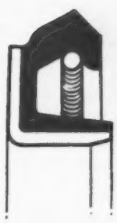
TYPE 11 P/B

As type 11 P but with the casing covered with rubber. Interchangeable with metal case seals.



TYPE 23 P

An external seal used when fitting conditions prevent the use of 11 P or 11 P/B seals.

TYPE 12 P

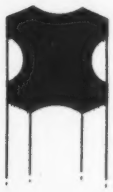
A rotary shaft seal used to separate two different fluids.

TYPE 13 P

Similar to type 11 P but without a spring on the second lip which serves as a dust excluder.

TYPE 31 P

A springless seal used as a wiper or dust excluder for rotating or reciprocating shafts.



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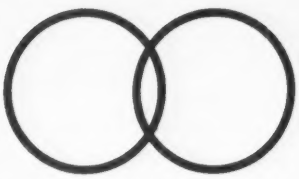

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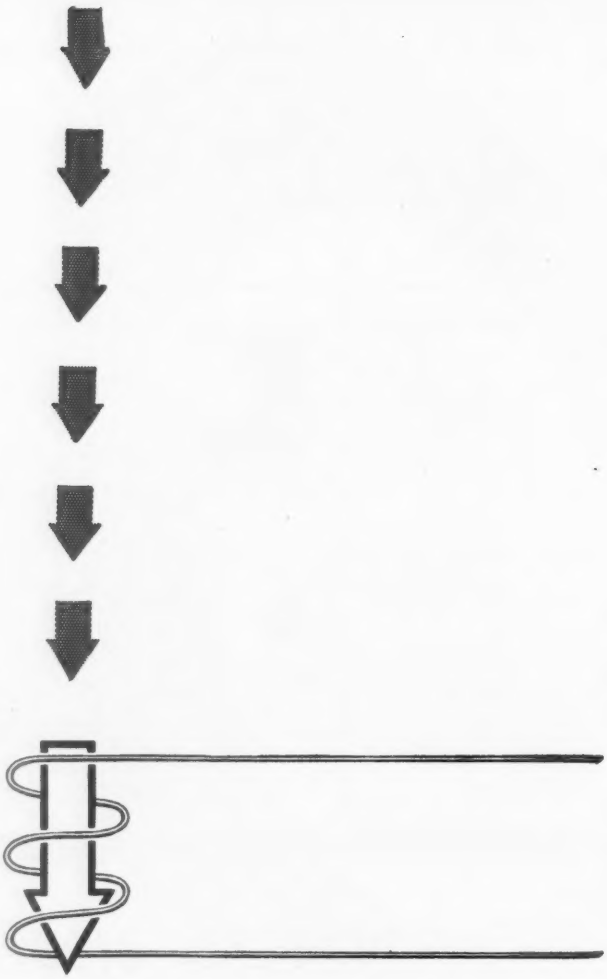

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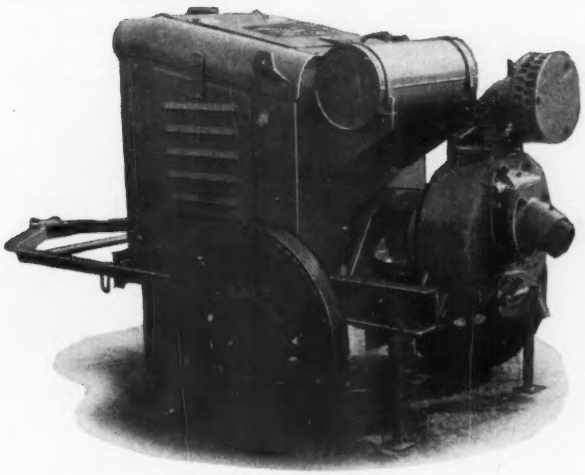

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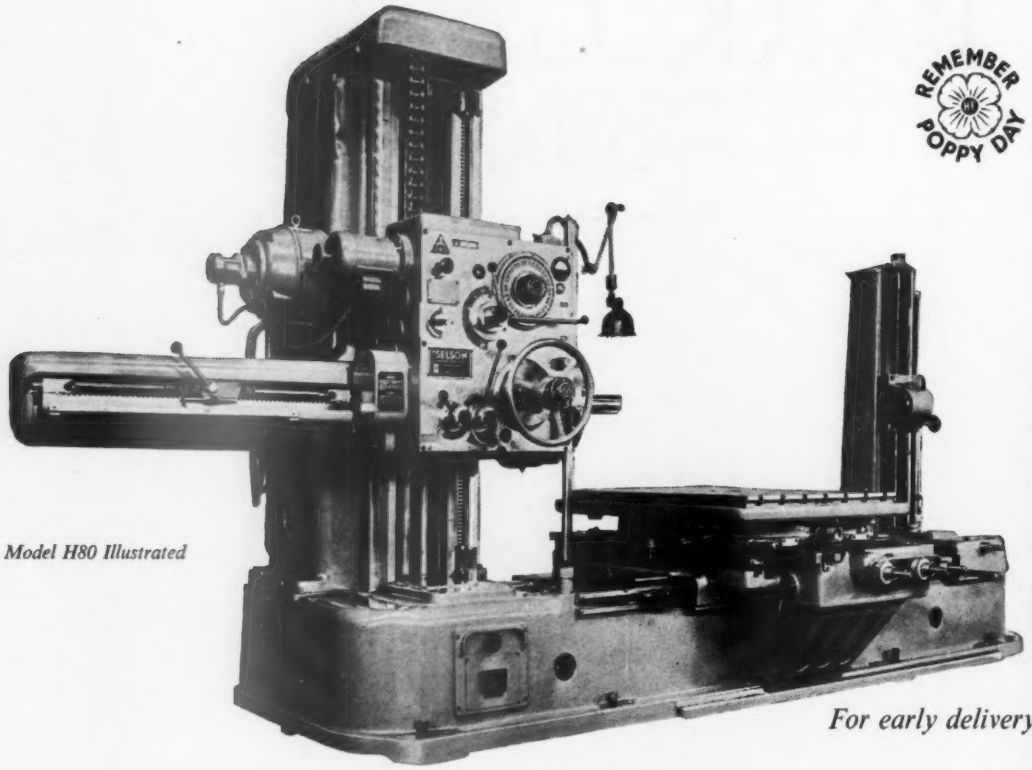

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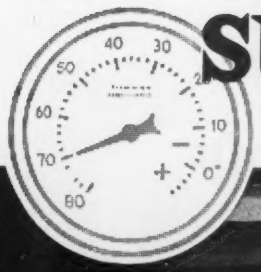

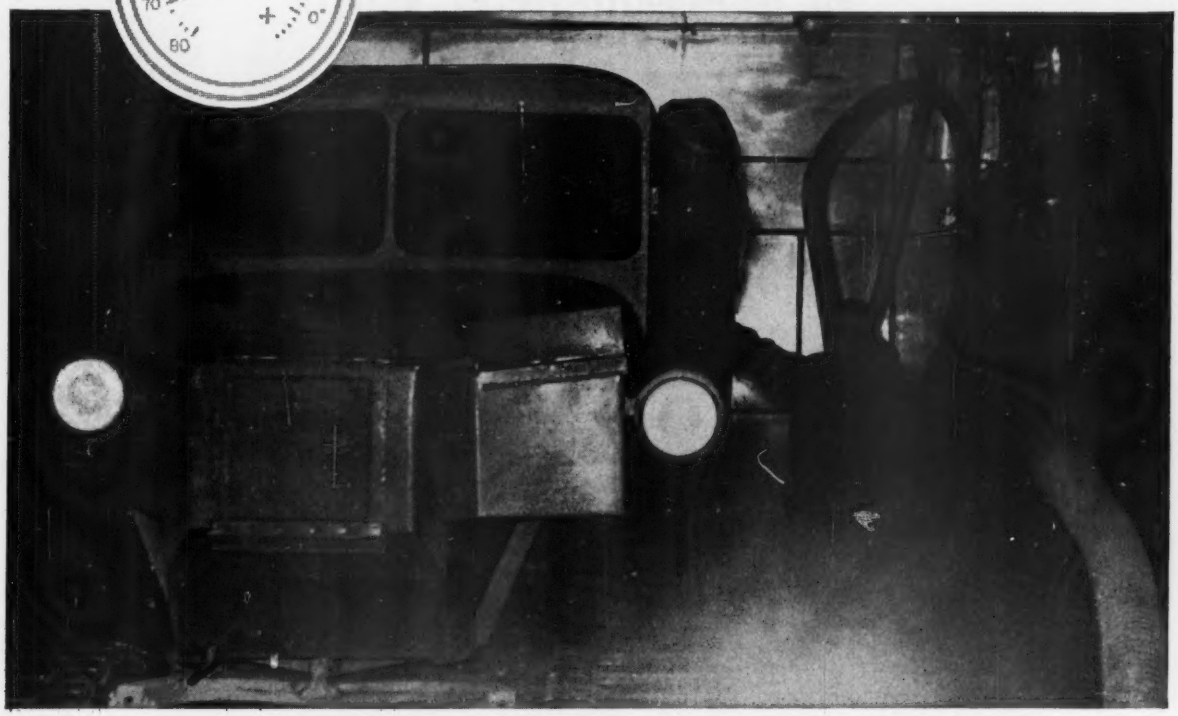

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- Access : One end of the chamber is removable to enable large equipment to be driven into the chamber.
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- Instrumentation can be conducted in the air lock which is maintained at a temperature of approximately 50°F, and facilities are available for inserting small bore oil or fuel lines and thermocouples from the air lock into the chamber.
- Air is normally dried to a frost point of −20°C to −25°C before reduction of temperature, so reducing the build-up of frost.
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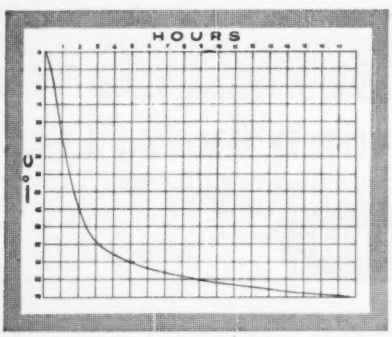

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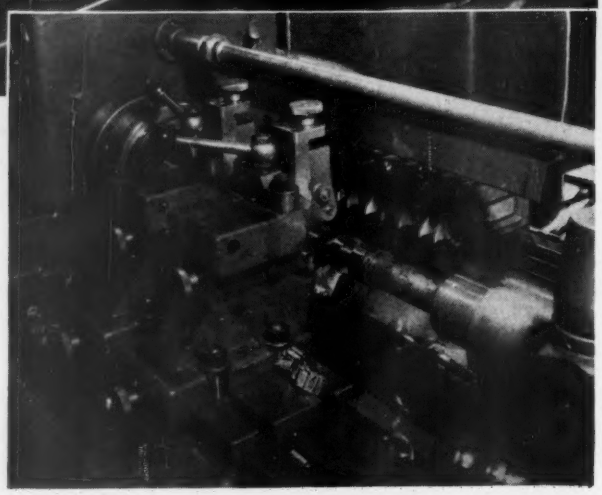
Help speed production of multi-diameter shafts



The " Maximatic " is shown tooled-up for turning steel camshafts. The operator's duties consist simply of unloading, loading and operating the main control lever. The tool slide carries out an automatic cycle of quick infeed, turn and quick return. All motors then stop ready for the component to be removed.

In the example shown, only the rear slide is employed for turning, but a combination of slides can be used for both turning and facing, each cross slide being independently operated by flat former plates, and adjustment can readily be made when a different component is required.

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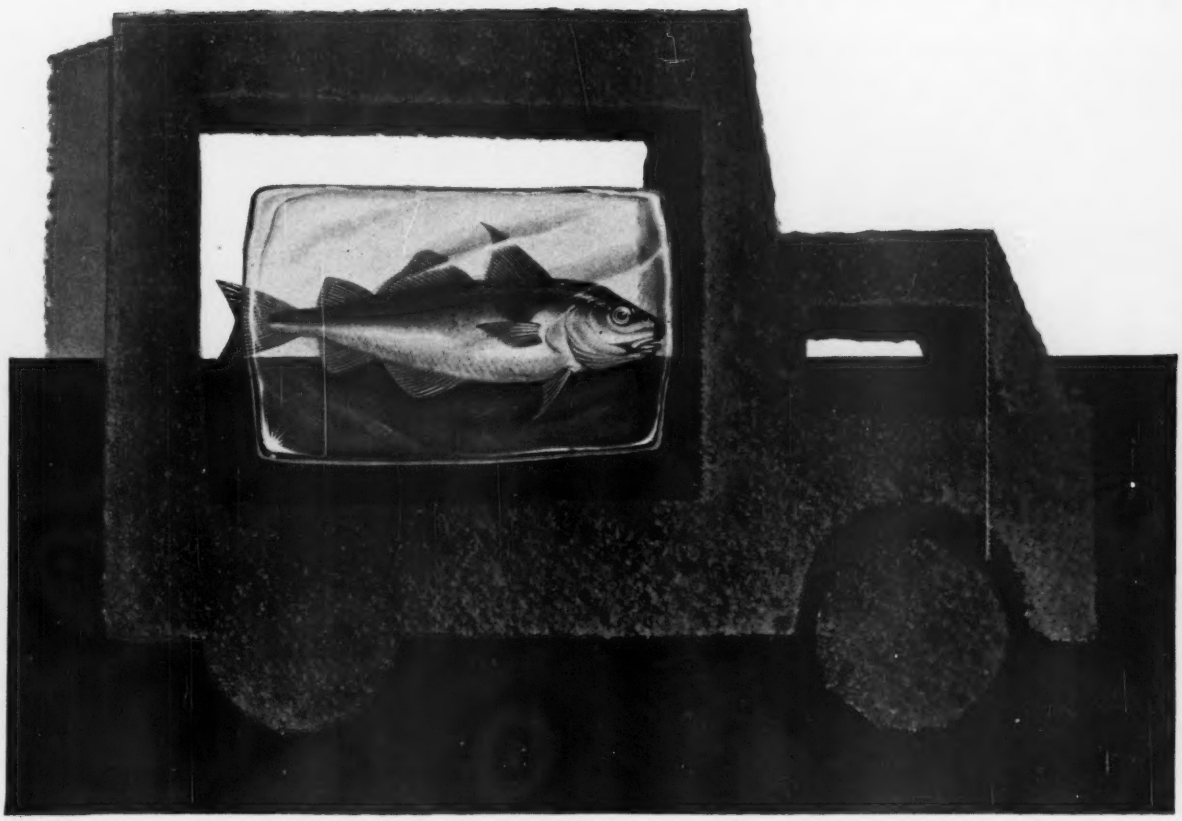

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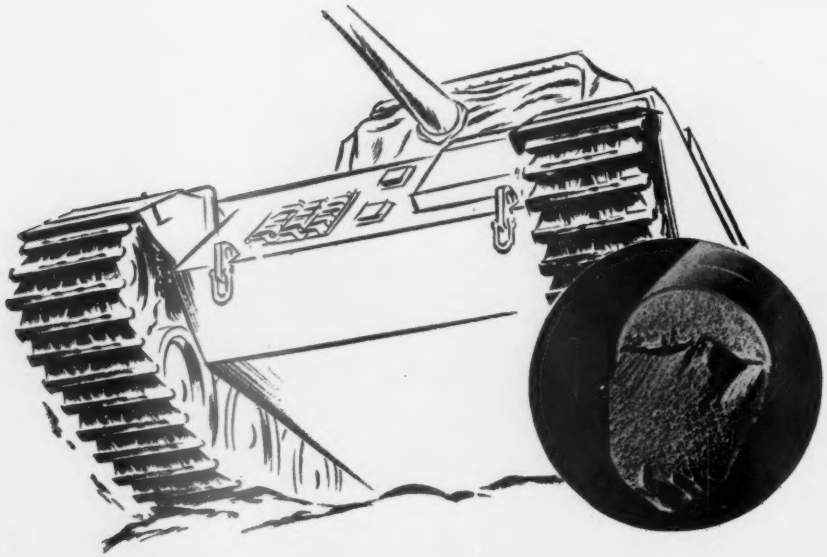

SURFACE HARDENING of tank track pins

Tank track pins are being surface hardened with a G.E.C. high frequency induction hardening machine at the Cardiff works of Curran Steels Ltd. The pins are hardened to a minimum of 54 Rockwell C and to depths varying from .025″ to .050″. Previously it was found difficult—and with one pin size impossible—to restrict the hardened surface to acceptable limits and all pins had subsequently to be straightened.

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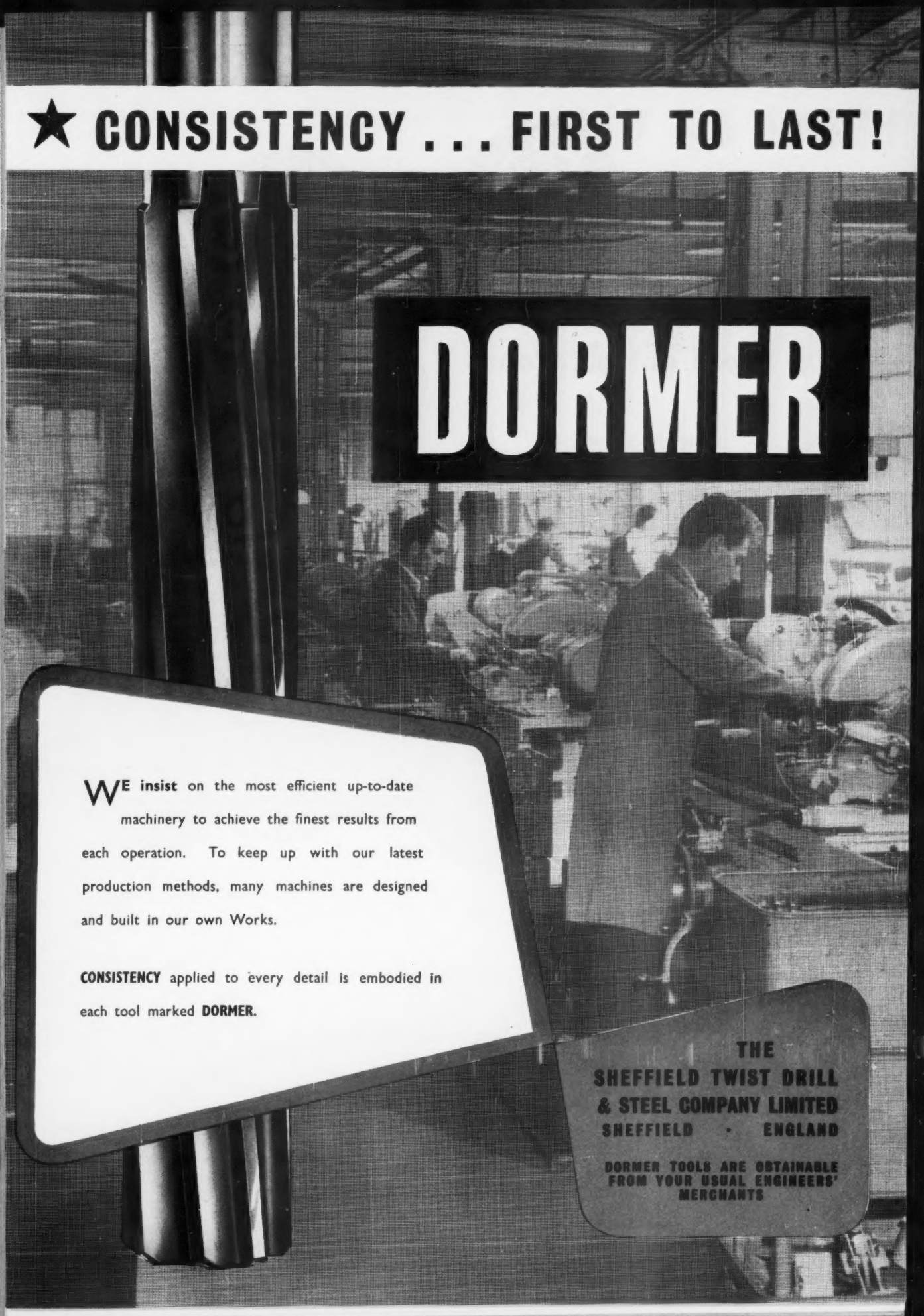
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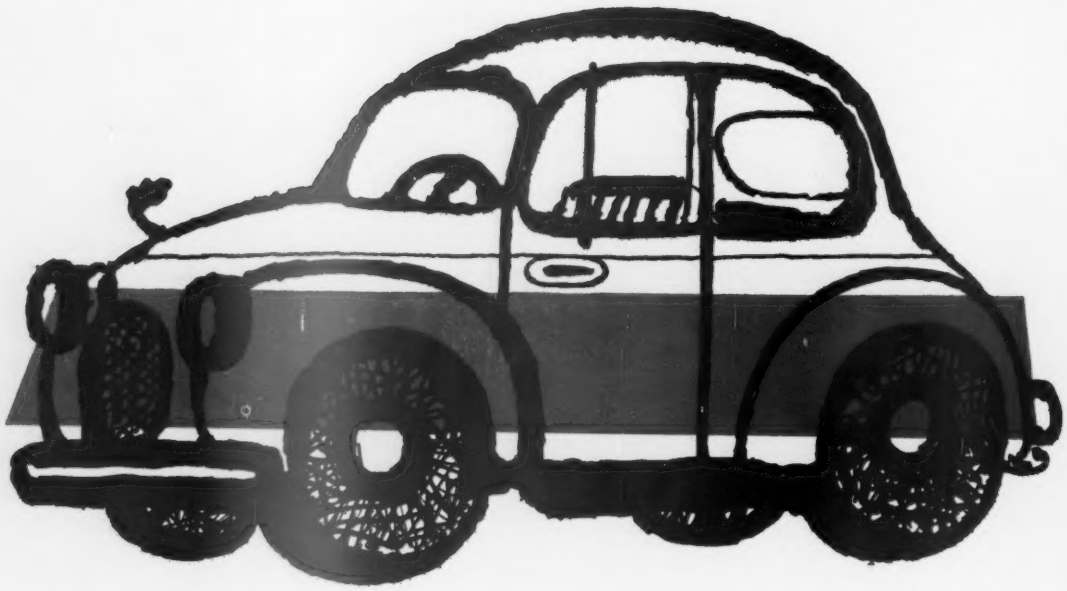

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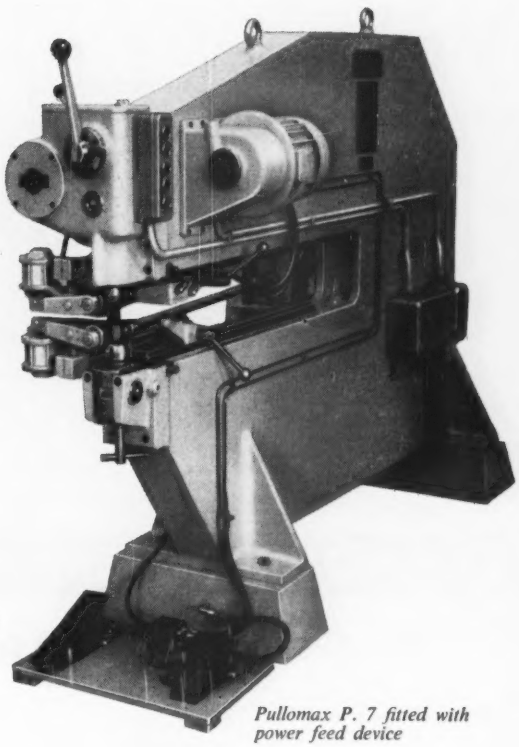

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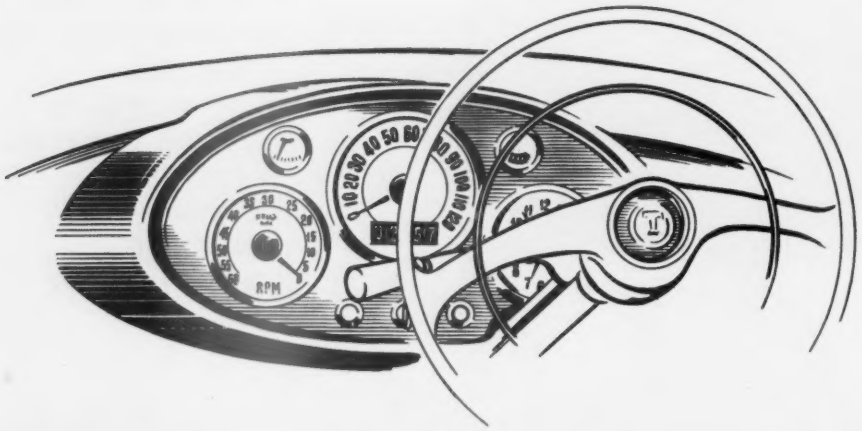

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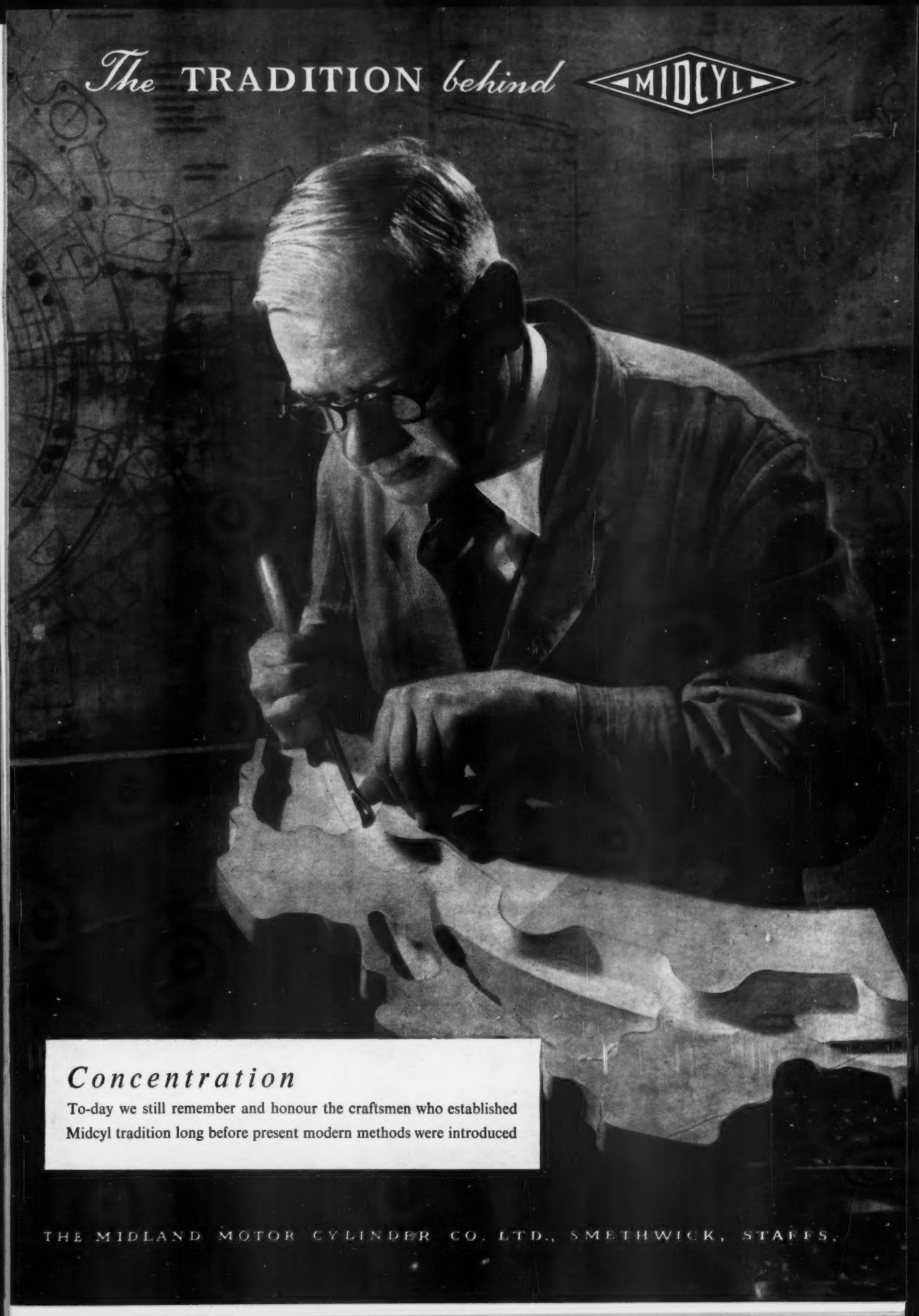
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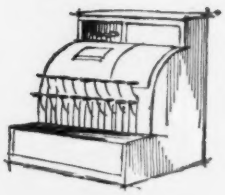

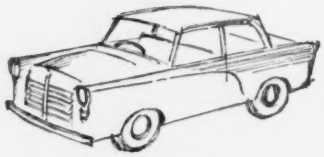

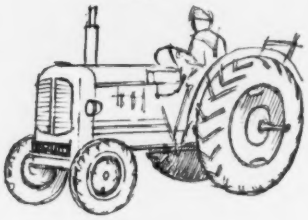

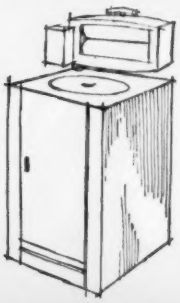

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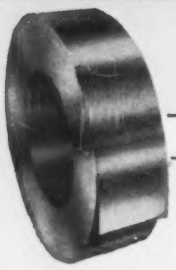

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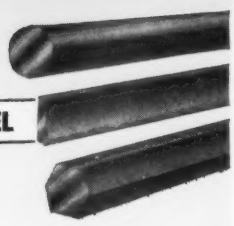

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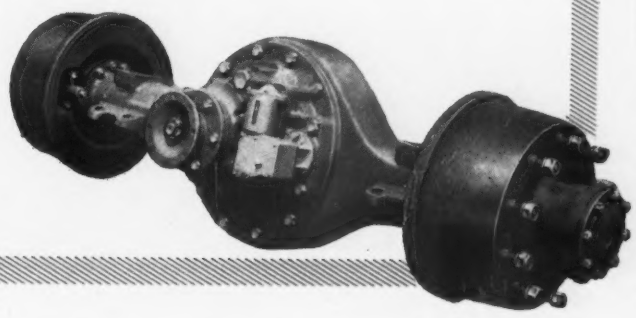

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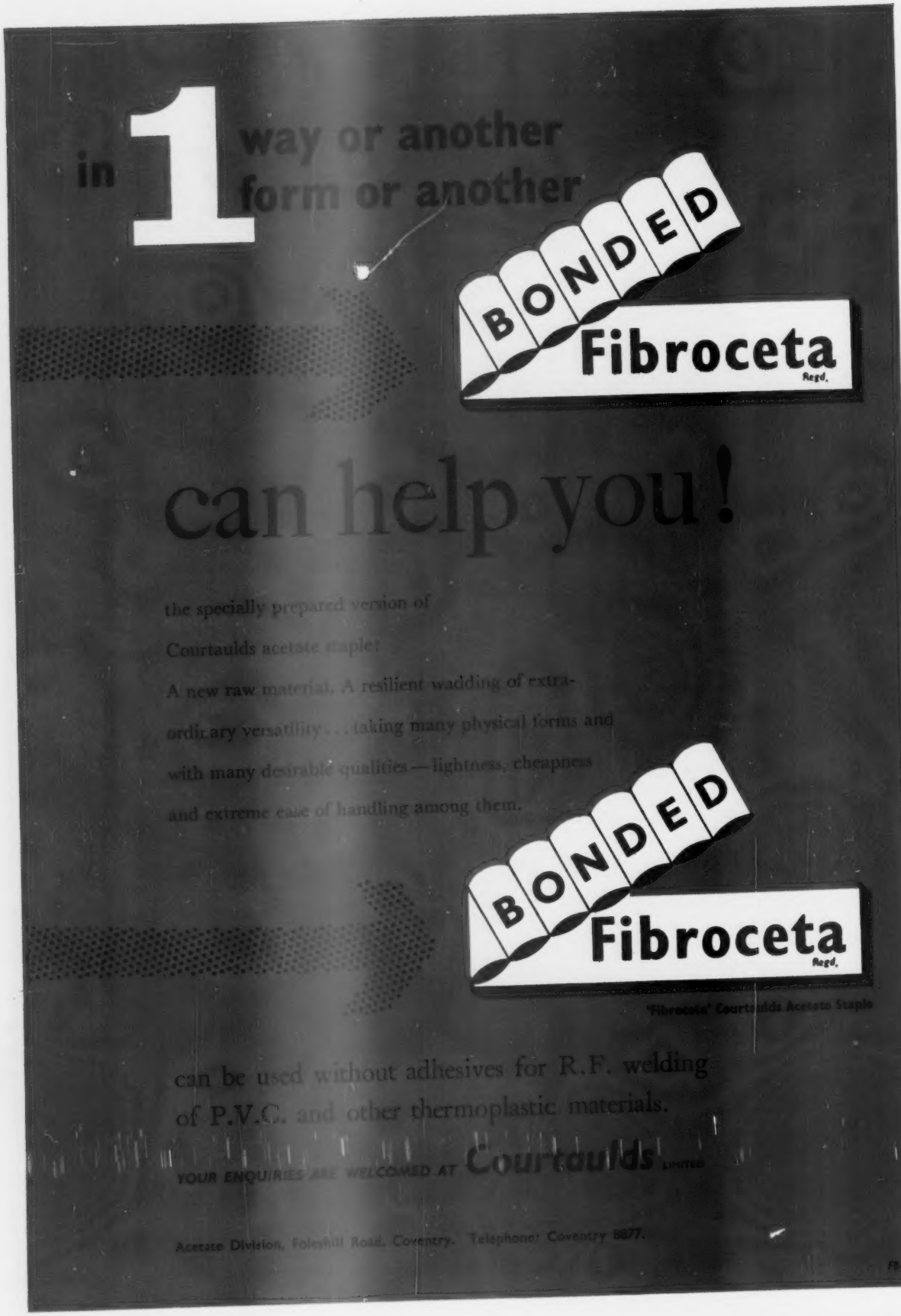
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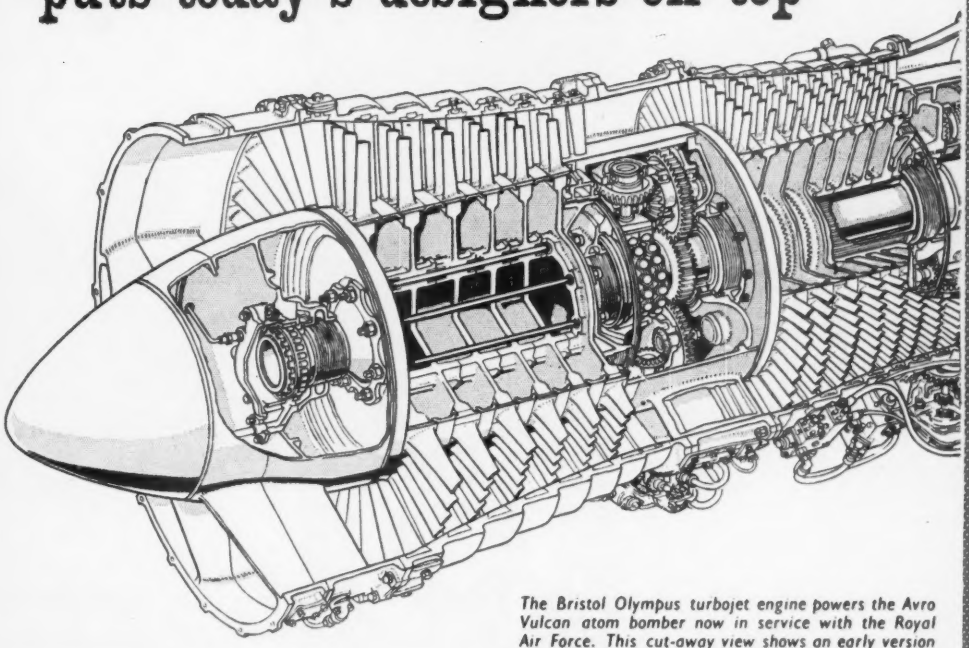

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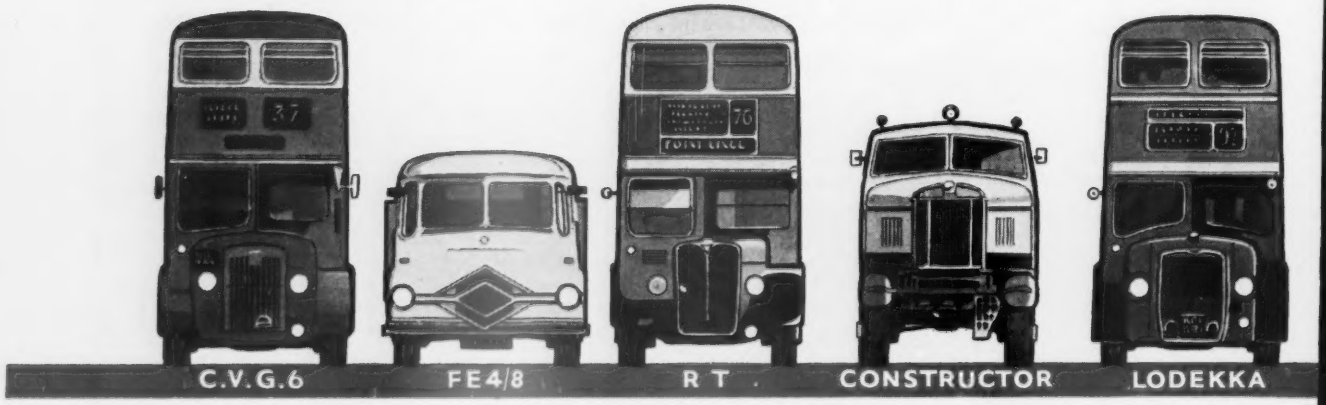

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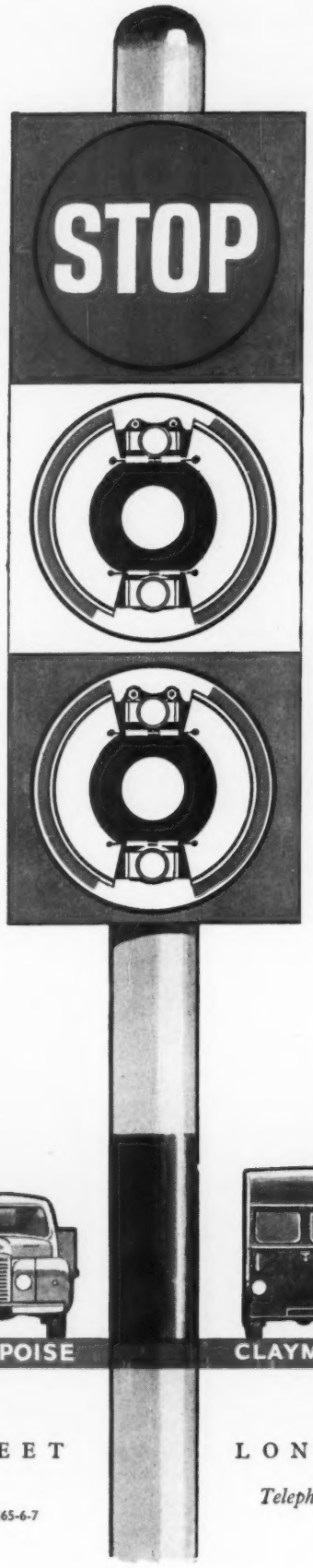

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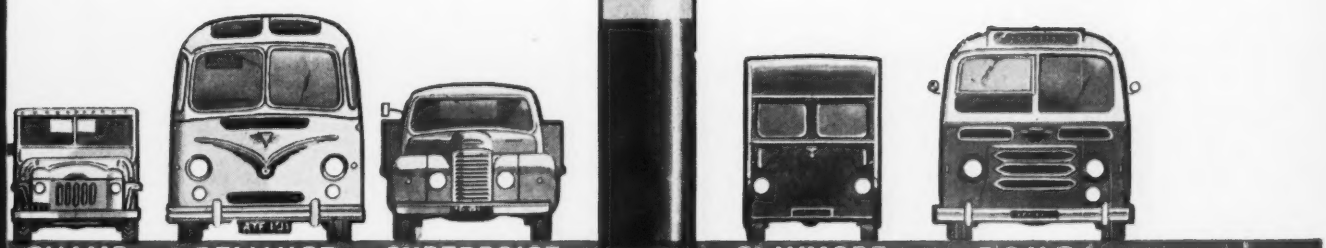

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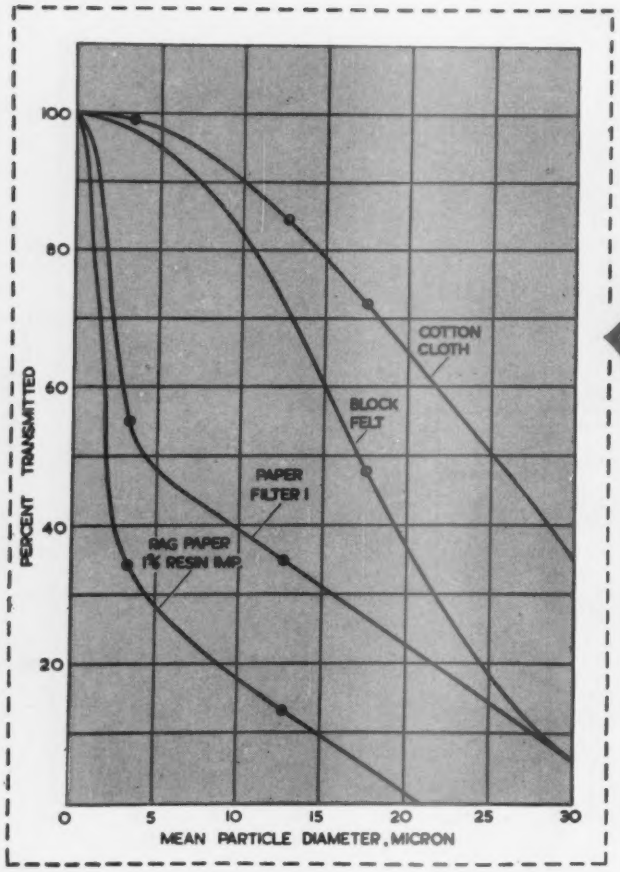
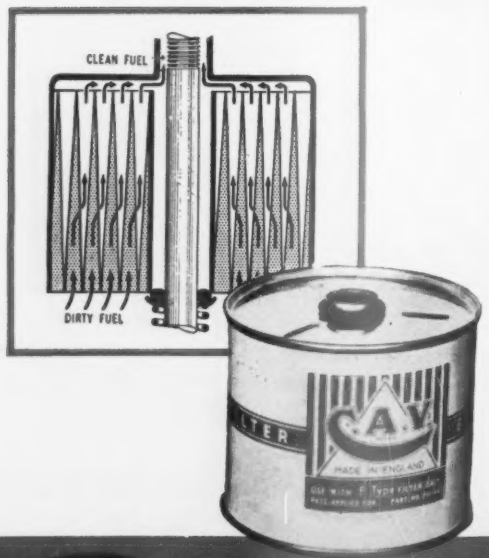


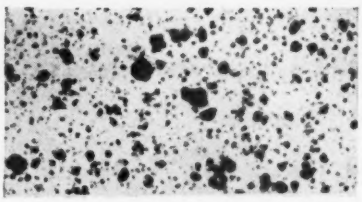
TESTS SHOW SUPERIORITY OF SPECIAL PAPER ELEMENT USED IN C.A.V. 'F' TYPE FUEL FILTER

Dust particles of any size carried in diesel fuel cause wear of injection equipment, but for a given weight of abrasive the wear is initially more rapid the smaller the particle size. The most damaging size of particle lies in the 6 to 12 micron range. Laboratory tests on the particle transmission of various filters were made, using specially prepared and graded dusts. The graph shows the specially treated paper used for the C.A.V. type 'F' filter to be by far the most effective material.

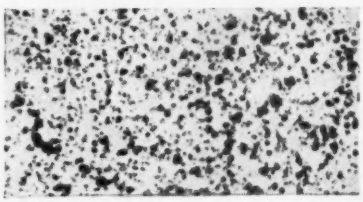
Further tests showed that the life of pump elements was increased by as much as 6 times over that obtained with a cloth filter.

The photomicrographs show fuel before filtering, after passing through a block felt filter, and after filtration through a C.A.V. paper element filter.





(1) *Photomicrograph (x 100) of fuel containing dust and dirt, before filtration.*



(2) *The same fuel as in (1) after passing through a block felt filter.*

(3) *The same fuel as in (1) after passing through a C.A.V. paper element filter.*

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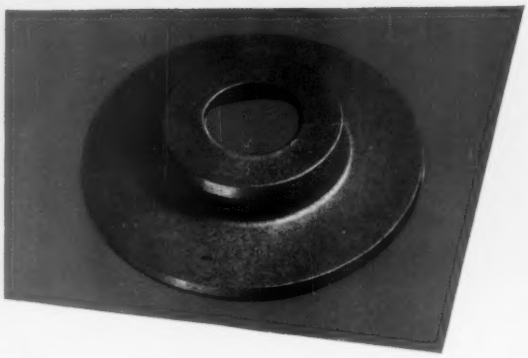

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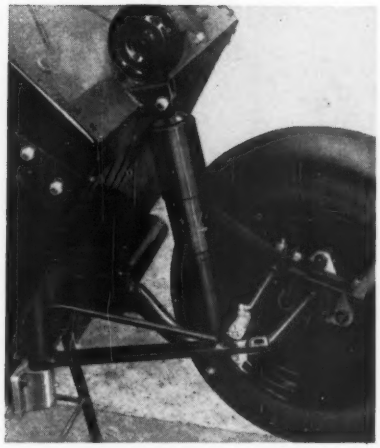
CONTENTS



ON THE HARPER INVALID'S CAR BRAMBER FLEXITOR UNITS ARE FITTED FOR THE SUSPENSION OF THE FRONT WHEEL, SHOWN ABOVE, AND ALSO FOR THE INDEPENDENTLY SPRUNG REAR WHEELS

425 Editorial *Scientists, Technologists, Technicians*

426 Renault Frégate Chassis *Part I: Transmission—A description of the design and some comments on its development*

442 Crankshaft Grinding *New developments in Newall machines and techniques*

444 The Trojan Senior *A diesel-engined 1-tonner now fitted with independent front wheel suspension*

446 Exhaust Gas Treatment *American investigations into methods for removing hydrocarbons*

449 Metal Cutting Research *Some activities of the Production Engineering Research Association*

456 Harper Mk. VI Invalid's Car *Petrol-driven and battery-driven models of a vehicle produced exclusively for physically handicapped drivers*

461 The Seating of Wet Clutches Dr. P. A. Vasavada *An analysis of the factors that determine the time required for a clutch of the wet disc type to engage fully*

463 Gear Lubricants *American developments for hypoid axle arrangements*

465 Current Patents *A review of recent automobile specifications*

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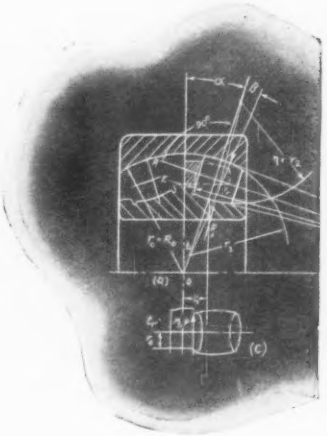

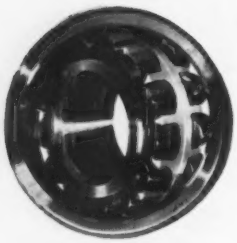

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F. C. Sheffield

DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Scientists, Technologists, Technicians

RECENTLY, and not for the first time, we heard it said that the British automobile industry is an industry of technicians rather than one of scientists and technologists. The remark was expressly aimed at the chassis manufacturers. If it be true, the technicians have much of which they may be proud. Nevertheless, it is worth while considering how far this pejorative statement is correct and what steps, if any, should be taken to alter the situation.

Unfortunately the terms scientist, technologist and technician do not have the same meanings for all people. For instance, whereas some would class all science graduates as scientists, others would define the word much more narrowly and confine its application only to those engaged in fundamental research. It is even more difficult to make a clear definition between the technologist and the technician. It may help to develop the argument if we make clear the meaning applied to the words in this context.

By scientist we mean a man who is competent to undertake research, either pure or applied. Such a person will, of course, be engaged on long-term projects rather than on day-to-day developments. For this a high standard of academic training is necessary. Over a very wide field, this class of work is best dealt with by some form of co-operative research of the type carried out by M.I.R.A. for the automobile industry. Despite the very valuable work that has been, and is being, done by M.I.R.A., there is still need for research by individual companies, but the available evidence suggests that chassis manufacturers are not so successful as other industries in attracting the right type of entrant. There is, of course, tremendous competition for scientists, and while the automobile industry can compete on equal terms so far as monetary rewards are concerned, it cannot offer such exciting projects as may be found in the newer fields such as atomic energy and guided missiles.

In many respects, the technologist is of greater immediate importance than the scientist. He, it is, who will be responsible for applying scientific developments, whether in design or in processes. Once again, a high standard of academic training is very desirable. Although the technologist may not be competent to initiate and carry through a research programme, he must be fully able to understand the results achieved by the scientist.

A great majority of science graduates are technologists rather than scientists. Their degrees show abilities of the order required to obtain the recently instituted Diploma of Technology. The graduate will not be as immediately useful to the industry as the man with the Dip. Tech., but since his academic training has covered a wider field, he can bring fresh viewpoints. It will be for the good of the industry if graduates are encouraged to undertake technological work. It is unfortunate that men with degrees seldom opt for work on the production side; almost without exception they choose design. This applies much more strongly in Great Britain than it does in the United States of America or in Germany. Here it is insufficiently understood that the production side of industry has problems that call for as clear analytical thinking as is required on the design side.

It is certain that most production technologists will be drawn from holders of Dip. Tech. but it would be a good thing if more graduates could be encouraged to turn their talents to the factory floor rather than to the drawing office or laboratory. Some universities are already trying to inculcate this idea. It will be interesting to see what success they have.

What the speaker quoted in the opening paragraph meant by technicians was not explained. In this category we should include draughtsmen, planning engineers, and the like. As the technologist translates the scientist's results into something industrially applicable, so the technicians translate the technologist's conception into a definite design or process.

In our opinion the case for a great deal more fundamental individual research within the industry is not very strong. In addition to the work carried out by M.I.R.A., there are such organizations as the Ricardo Laboratories, the Weslake Laboratories and the Fulmer Research Institute for sponsored projects. A full use of these, and similar, organizations will probably be of greater benefit to the industry than an attempt by each individual firm to extend its facilities for long-term research. What is required is an increase in the number of technologists competent to apply scientific developments.

Renault Frégate Chassis

Part I: Transmission—A Description of the Design and Some Comments on its Development

GOOD road-holding qualities are an outstanding feature of the Renault Frégate. They have been obtained by the employment of independent rear suspension, in conjunction with a conventional double transverse link type front suspension system. There can be little doubt that independent rear suspension has come to stay in France, since motorists there have now grown accustomed to it and are unlikely to accept anything inferior for this type of car. The fact that smaller models, such as the Dauphine and other well known Continental cars, also have independent rear suspension makes it even more obvious that models to be introduced later by this and other companies will retain the independent rear suspension feature. In a country where most towns have pavé road surfaces, it is not surprising that in road vehicle design more emphasis is placed on suspension than is the case, for example, in Great Britain and America, where good tar macadam surfaces are the general rule in the towns, as well as on the long distance routes used by intercity traffic.

When the Renault Frégate design was conceived, the manufacturers realized that its good road-holding qualities would inevitably lead to higher speeds and that its braking capacity would, therefore, have to be correspondingly good. All these features added to the cost of the vehicle, but the overall result was so good that the demand has been maintained at a high level over a long period and, by virtue of the large quantities produced, it has been possible to offset the cost of the refinements that were introduced into the design.

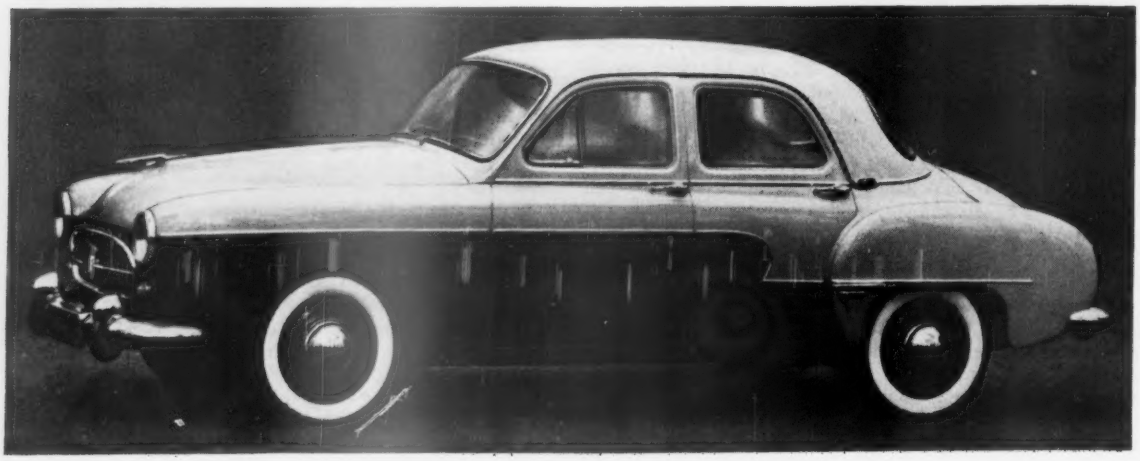
An unusually long wheelbase, 9 ft 2¼ in, has been adopted. It is of interest to compare this dimension with that of some other cars of similar size. The wheelbase of its closest rival, the Ford Versailles, is 8 ft 6 in; that of some of the British cars is as follows: Standard Vanguard III 8 ft 6 in, and Vanguard II 7 ft 10 in; Austin A70 8 ft 3 in, and A95 8 ft 9¾ in; Daimler Conquest 8 ft 8 in; Rover 60 9 ft 3 in; Vauxhall Velox 8 ft 7 in; the latest Ford Zephyr 8 ft 11 in; and Consul 8 ft 8½ in, while the earlier Consul model was 8 ft 4 in.

A long wheelbase is desirable, of course, to obtain stability and to give as much room as possible for the passengers. It is practicable in this car mainly because the final drive unit is secured to the vehicle structure instead of oscillating with the axle. This feature has made possible the incorporation of an intermediate propeller shaft bearing without any danger of vibration problems due to variations in the angularity of the shafts. Considerable care and thought have been devoted to the design of the mounting for the intermediate bearing.

The engine is well forward, and the rear end of its sump is shaped to clear the cross member that carries the front suspension. By virtue of the forward position of the engine and suspension assemblies, and the arrangement of the steering rods above the clutch bellhousing, the toeboard pressing is only slightly dished locally to clear the gearbox, and it is not greatly obstructed by the intrusion of the wheel arches.

In the earlier versions of this model, there was no need for a propeller shaft tunnel; this was because of the low propeller shaft line obtained by taking the drive from the rear end of the layshaft, through the intermediate bearing, to a fixed final drive unit. There was, however, a slight bulge in the floor over the intermediate bearing assembly. The forward portion of the propeller shaft was almost horizontal and the axes of the shafts in front and behind it were inclined at approximately equal, but opposite, angles to it. When subsequently the decision was taken to employ a gearbox with direct instead of overdrive top speed, and the drive therefore had to be taken from the rear end of the gearbox mainshaft, a small tunnel had to be incorporated in the floor, forward

Among the noteworthy features of the Renault Frégate is the fact that the rear wings can be readily removed for the repair of accidental damage



of the intermediate bearing. Also a hypoid axle was substituted for the simple, spiral bevel unit, so that a favourable transmission line could be obtained.

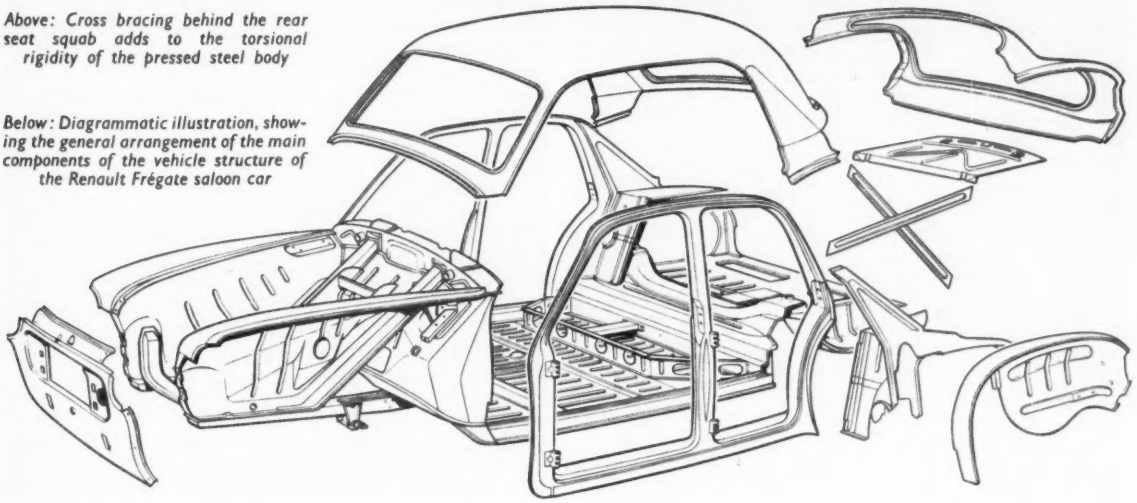
The axis of the engine and gearbox is now inclined upwards at an angle of 3 deg 9 min, from the front flexible joint, while the forward portion of the propeller shaft is inclined at 0 deg 36 min downwards from the horizontal. The rear portion of the propeller shaft and the hypoid pinion shaft of the final drive unit are in line, and their common axis slopes down at an angle of 2 deg 5 min towards the intermediate bearing. In the original layout, the engine was at the same angle as in the later version, the front portion of the propeller shaft was set 1 deg down from the horizontal, the rear portion at 2½ deg up, and the final drive unit 2½ deg up.

A short while ago, it was announced that in 1958 the de luxe version of the Frégate, known as the Grand Pavois, will be offered with a two-pedal, semi-automatic transmission, called Transfluide. This system consists of a hydraulic torque converter, a single-dry-plate clutch with electromagnetic control, and a three-speed and reverse synchromesh gearbox.

Some weight saving has been effected by employing a

Above: Cross bracing behind the rear seat squab adds to the torsional rigidity of the pressed steel body

Below: Diagrammatic illustration, showing the general arrangement of the main components of the vehicle structure of the Renault Frégate saloon car



trailing link type rear suspension layout. The links are pivoted on to a cross member below the rear seat squab. This has enabled a relatively light structure to be used to the rear of the squab and cross member, and, of course, the coil springs employed are also much lighter than semi-elliptic ones.

The forward face of the front seat squab is almost exactly mid-way between the front and rear wheel axes, while the rear seat squab is just forward of the rear wheel arches. Thus, the full width of the interior has been made available for the passengers, and a good weight distribution obtained. In fact, the weight distribution figures are: dry, 699 kg front and 507 kg rear; in running order, 720 kg front and 595 kg rear; fully laden, 860 kg front and 925 kg rear.

All the mechanical components are arranged more or less symmetrically in front of and behind the forward portion of the propeller shaft, which has Layrub flexible couplings at each end. The mounting of the engine and gearbox was described in the article on the Renault 2·1 litre engine, published in the September 1957 issue of *Automobile Engineer*.

Another interesting mounting system is that for the final drive unit. The problem with this type of layout is to provide for the reaction of the final drive and input torques, as well as for the isolation of the relatively high frequency vibrations of the gearing system and the lower frequency vibrations transmitted through the propeller shaft to the unit. Since the final drive, or output, torque is relatively high, the mountings have to be wide apart as viewed in side elevation. In this application, the rear mountings are 138 mm behind the axis of the differential gears, while the front mountings are 750 mm forward of this axis. The wide spacing of the mountings enables soft rubbers to be employed to cater for vibration absorption. In all, six rubbers are used. They are carried on two cross members, four mountings being fitted on the rear cross member and two on the front one. Each mounting comprises two circular rubbers, one above and the other below the cross member. They are each spigoted into the clearance hole for the bolt that secures them. In other words, the arrangement is similar to that commonly employed for the attachment of telescopic shock absorbers, the rubber spigots ensuring that there is no metal-to-metal contact between the bolt and the cross member.

Two brackets carry the front mountings. They are bolted, one to each side of the nose piece of the final drive casing,

View of the rear end of the body shell, showing how the wheel arches and rear decking are joined to the underfloor structure to add to the strength and stiffness of the vehicle and to give a large rectangular-shaped boot for the luggage

and they extend forwards on each side of the propeller shaft. Their front ends are bolted to the cross member already mentioned. The rubber mountings are fitted one at each end of this cross member and are bolted up to the main longitudinal members of the vehicle structure, below the main cross member under the rear seat squab. This cross member also carries the pivots of the trailing links of the rear suspension.

At the rear, the arrangement is slightly more complicated. One pair of rubber mountings is interposed between the cross member and lugs extending rearwards from the final drive unit, and another pair is fitted between the ends of the cross member and the main longitudinal members of the vehicle structure. The intermediate pair of mountings, which are 180 mm apart, are smaller than the outer pair of mountings, which are 435 mm apart. The distance between the front two mountings is 520 mm.

Vehicle Structure

The main components of the underfloor structure are two longitudinal, box section members, extending parallel from front to rear of the vehicle. They are broken only at the cross member under the rear seat squab. In front of this cross member, the axes of the longitudinal members are 520 mm apart and behind it they are 435 mm apart. Throughout the whole of their length they are formed by inverted top hat sections, the width of which is 50 mm plus the width of the flanges by means of which they are welded to the body floor.

At the front, where they form cantilevers to carry the engine and front suspension, the top hat sections are each closed by a plate welded on the flanges. This cantilever portion is 62·5 mm deep at its front end, becoming 82·5 mm under the dash; thence, over a length of 630 mm to the point where it joins the cross member just behind the intermediate bearing, it tapers to a depth of 45 mm. Between the intermediate bearing and the heelboard it increases in depth progressively to 57·5 mm at the rear. The rearmost portion forms a gusset between the heelboard and the boot floor. It is 113·5 mm deep at the heelboard and tapers to a depth of 25 mm over the rear axle.

There are, of course, two box section sills, one on each side, and a number of cross members. At the extreme front end, there is a box section cross member under the radiator. Immediately behind it, the engine mounting brackets are fitted on the side members. A tubular cross member carries the suspension units. It has swan-neck forgings welded on each end: these forgings fit under the longitudinal members and are each secured by two 14 mm diameter bolts, assembled from above, through the box section side members. Reinforcement plates are spot welded locally to the inner faces of the top and bottom walls of the box sections, and distance tubes, through which the bolts pass, are interposed between them. These bolts also hold down a thick forged steel angle bracket on top of the longitudinal members, the other arm of this angle bracket being secured to the vertical part of the swan-neck fitting that carries the suspension assembly. Further to the rear, each longitudinal member has bolted to it a bracket to carry the rear mounting of the engine and gearbox unit.

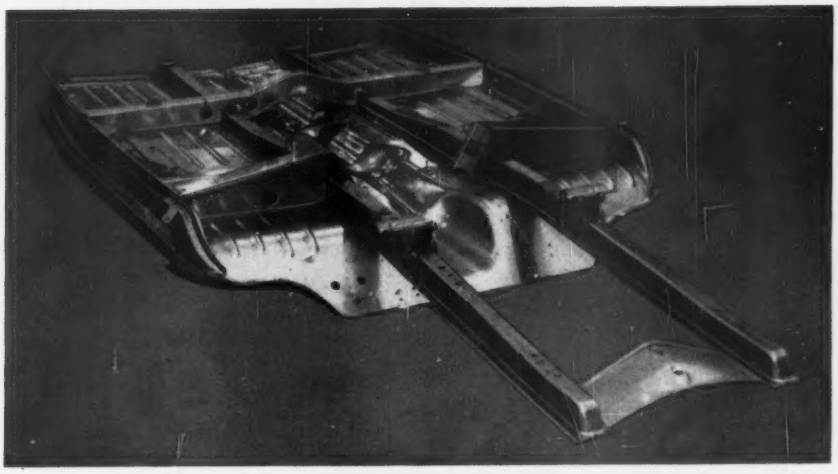
The remainder of the cross members are under the floor of the body, and are of inverted top hat section, welded by their flanges to the floor. The front one is under the toeboard, but is broken to accommodate the main longitudinal members. The centre piece is welded between the longitudinals and is arched under the propeller shaft, while the two outer pieces extend to the body sills. This cross member is 80 mm wide, excluding its flanges, and is 145 mm deep where it joins the longitudinal members. The cross member immediately behind the propeller shaft intermediate bearing is 65 mm wide and 80 mm deep at the centre, where it is arched under the propeller shaft.

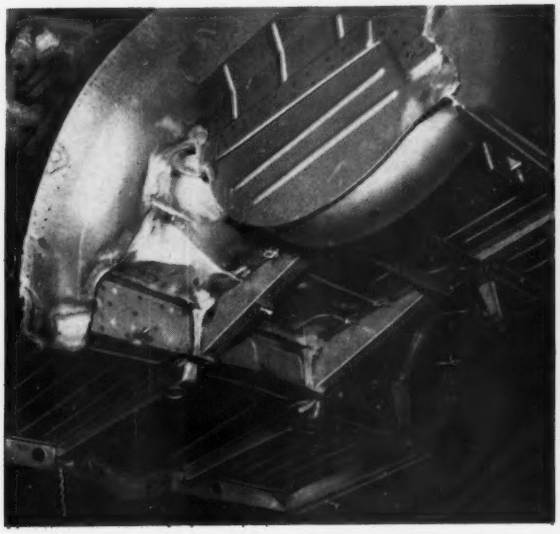
Under the rear seat squab, there is a box section cross member. A stepped, vertical panel supports the rear edge of the seat pan, and a vertical closing plate behind it helps to support the forward edge of the boot floor. These two components form the front and rear walls of the box section. The cross member is about 167 mm deep where it carries the pivots for the trailing links of the rear suspension and the width of its cross section is 70 mm.

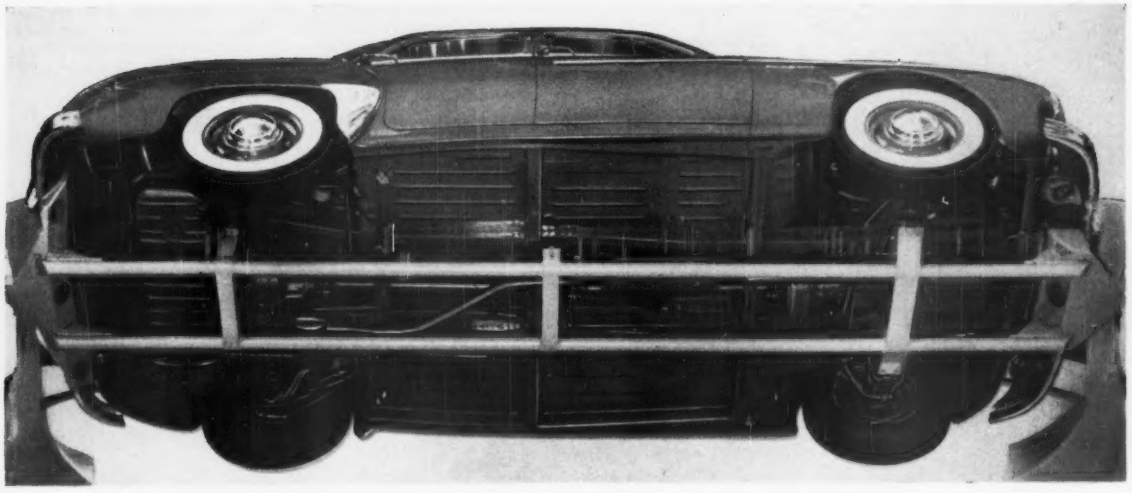
Another large top hat section member is spot welded on top of the boot floor between the wheel arches. Its ends are shaped to carry the spring pans and the mounting brackets for the upper ends of the telescopic shock absorbers. At the centre this member is 65 mm deep by 100 mm wide, excluding its flange. It is of 1½ mm thick sheet steel. This crosss member, together with that under the rear seat squab, takes all the suspension loads.

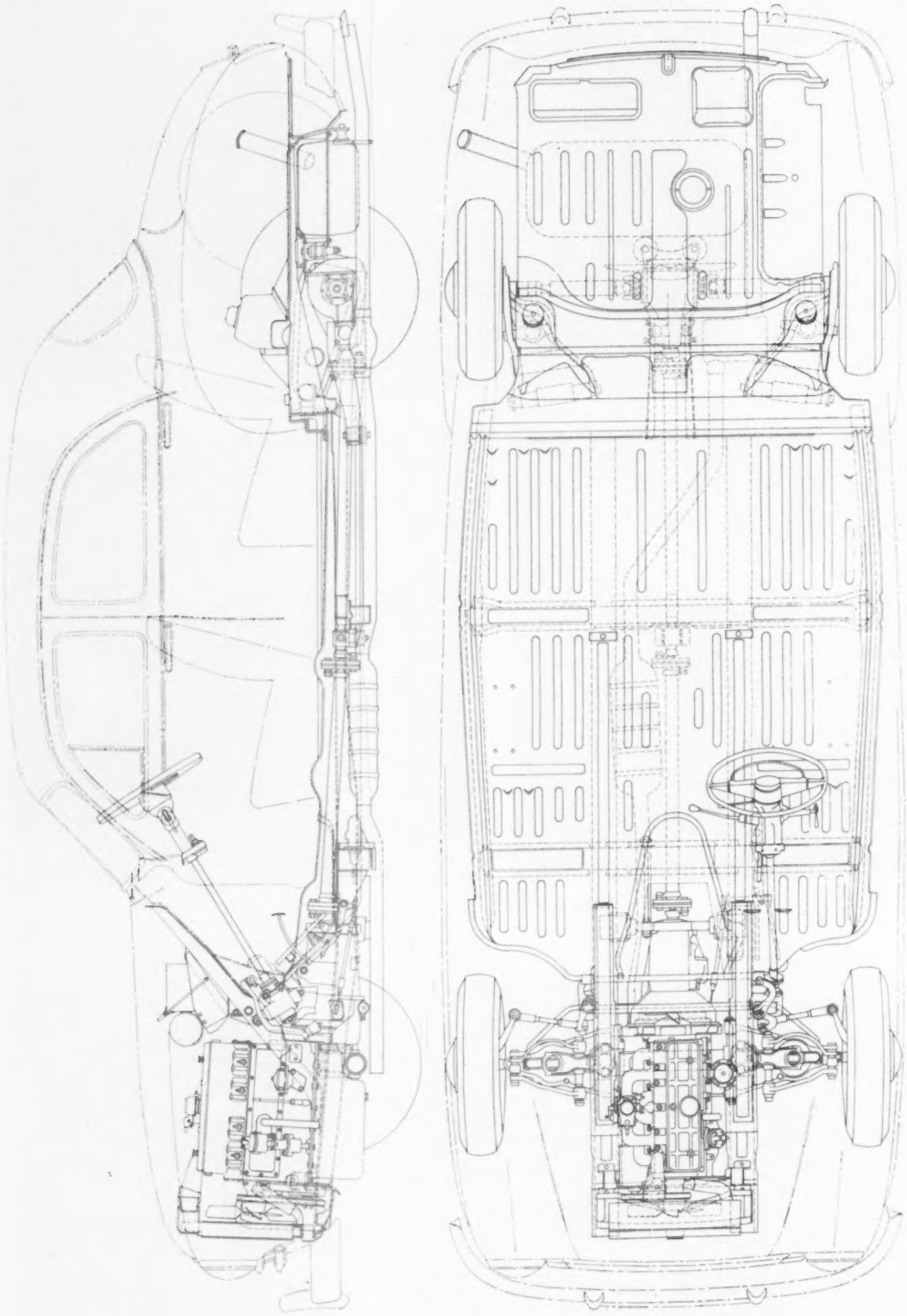
Another cross member is formed by a closing plate between the lower edge of the outer panel and the rear edge of the boot floor. To carry the rear mountings of the final drive unit, a small cross member bridges the space between the

The illustrations on this page show different aspects of the body structure, including the basic underfloor frame, the various mounting points for the main mechanical components

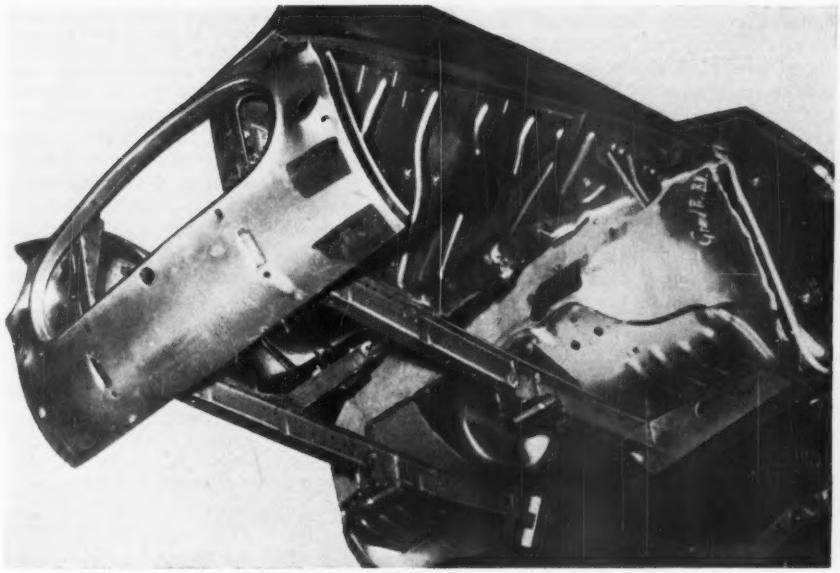








GENERAL ARRANGEMENT OF THE RENAULT FRÉGATE CHASSIS



Box section, cantilever members extend forward from under the toeboard to carry the front suspension cross member and the engine mountings. Additional support for these members is furnished by bracing struts between the scuttle and the dash, the wheel valance panels, and the front skirt panel

longitudinal members. All the floor panels are stiffened between the cross members by flutes parallel to the longitudinal axis of the vehicle. This arrangement not only adds to the overall rigidity of the structure, but also helps to prevent the panels from drumming and thus reduces the noise level.

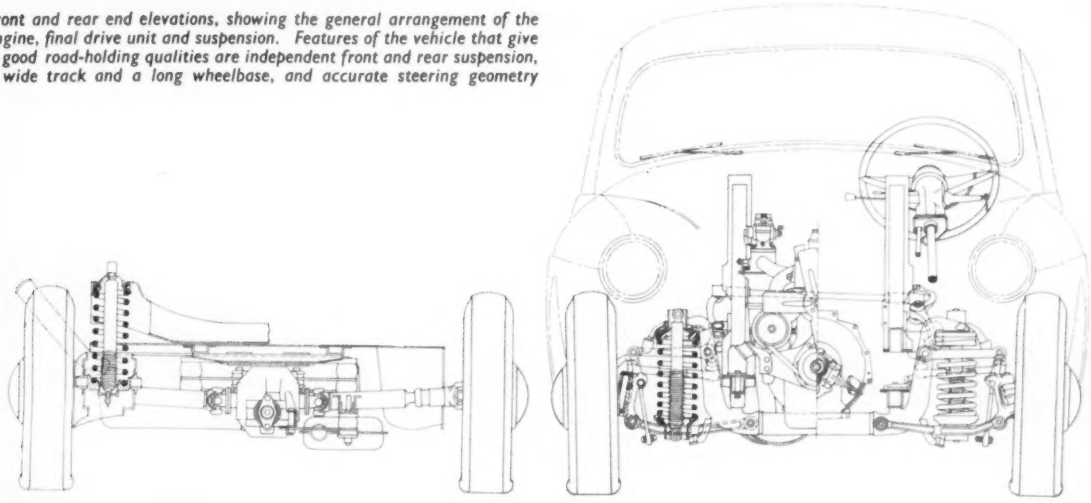
The frame is integral with and supported by the body panelling above it. For example, the wing valances at the front are welded to the cantilever members and their forward edges are welded to the grille surround panel. Additional support for the cantilever members is afforded by two box section struts, one on each side, extending between a point just forward of the suspension cross member and the scuttle. These struts also carry the steering box and idler lever assembly. Their width is the same as that of the longitudinal members and their depth is 55 mm over most of their length. They are of top hat section closed by a plate welded below their flanges. This plate extends only between the lower end and the point where the strut passes over the sloping dash panel. Above this point, the flanges of the top hat section are spot welded to the dash panel and to the scuttle, to spread the load over a large number of welds and over a big area of metal. A transverse cross member of triangular section, is formed by the scuttle in the following manner. The upper edge of the sloping dash panel is spot welded to the edge of the opening for the windscreen, and a vertical panel connects the front edge of the scuttle and the sloping dash panel.

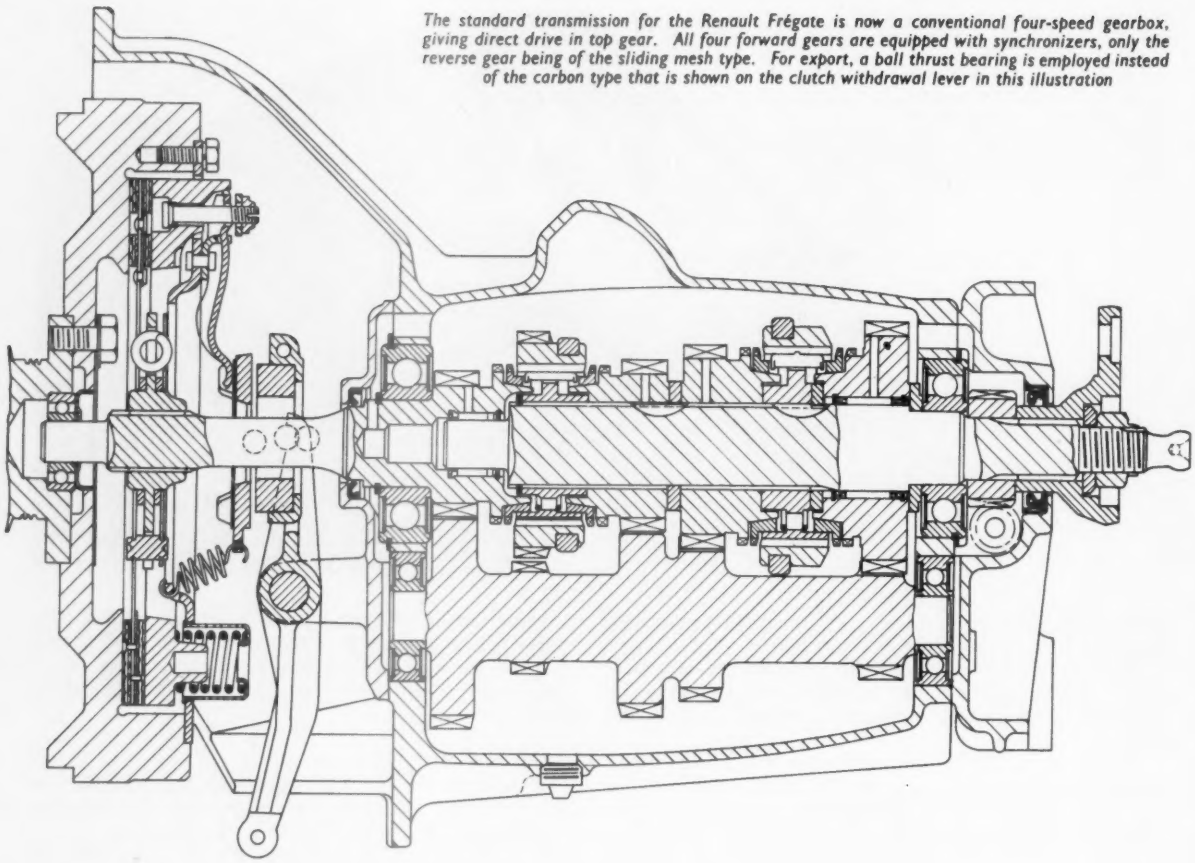
Clutch and gearbox

A Ferodo PKS C13, single-dry-plate clutch transmits the drive from the 2·1 litre engine to the gearbox. The diameter of the friction plate is 214 mm and the area of the friction lining is 5,290 mm². Coil springs are interposed between the hub and the centre plate to absorb shock and vibrations. There are nine presser springs, each of which exerts a load of 42 kg. A graphite thrust bearing is employed, except in the export models, which have a ball thrust bearing. The movement of the presser plate between its engaged and disengaged positions is 11 mm and, to compensate for wear, a further 8·5 mm can be obtained by adjustment of the control.

Originally, this vehicle was equipped with a gearbox in which the top gear was an overdrive. This layout has now been abandoned and a more conventional gearbox with a direct drive top gear has been adopted. Both boxes have

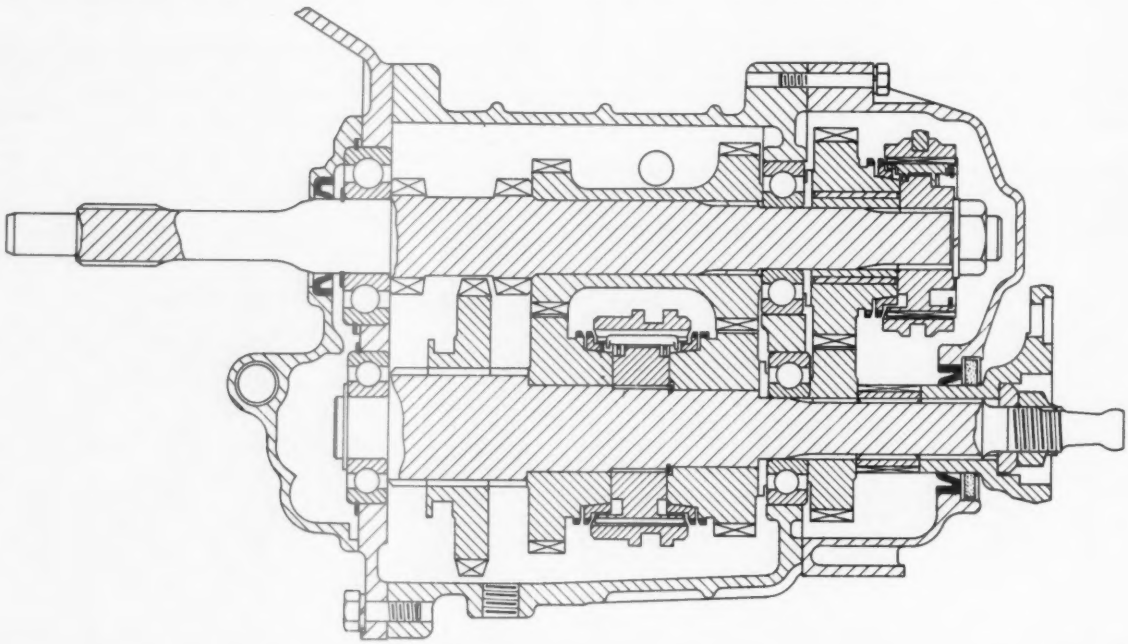
Front and rear end elevations, showing the general arrangement of the engine, final drive unit and suspension. Features of the vehicle that give it good road-holding qualities are independent front and rear suspension, a wide track and a long wheelbase, and accurate steering geometry





The standard transmission for the Renault Frégate is now a conventional four-speed gearbox, giving direct drive in top gear. All four forward gears are equipped with synchronizers, only the reverse gear being of the sliding mesh type. For export, a ball thrust bearing is employed instead of the carbon type that is shown on the clutch withdrawal lever in this illustration

The gearbox illustrated below is fitted to the earlier versions of the Renault Frégate. It has an overdrive, indirect top gear; the bottom and reverse gears are of the sliding mesh type, and the other three are single-helical, constant mesh gears, equipped with synchronizers



four forward speeds and one reverse. An accompanying illustration shows the layout of the older type of gearbox.

A noteworthy feature of the original box is its compactness; the overall length of the casing, including the rear cover, is about 10¼ in. Also, since the gears are all indirect and the propeller shaft is, therefore, connected to the layshaft, there is no need for a propeller shaft tunnel in the body floor. However, it suffers from the disadvantage that it is more noisy in top gear than is the direct drive gearbox.

In the older box, the top speed gear pair is overhung behind the rear bearings. This arrangement has the advantage that both the top and the third speed gears are as close as possible to a shaft support bearing. The provision of adequate support for these gears is, of course, important not only because they are the two speeds that are most used, but also because they are both indirect and therefore, liable to be unacceptably noisy if they do not mesh accurately.

With the new box, the steering column gear shift control is still employed, and the selector rods are housed in a cover bolted to the side of the box. This selector rod assembly is forward of the toeboard and, therefore, does not encroach upon the space for the feet of the driver and the front passenger. The overall width of the casting is 210 mm, which is about 8¼ in. Its length between the centres of the front and rear bearings is 222 mm, approximately 8¾ in. It might be assumed from this latter figure that the new box is shorter than the old one; however, this is not so, because the length between the forward face of the front wall and the rear face of the housing for the tail-end seal is 273 mm, 10¾ in.

Since the earlier gearbox is not now in production, the description that follows will be confined to the new one. An iron casting forms the integral gearbox and bellhousing. A bolted-on cover retains the two bearings at that end of the box, and houses the Paulstra No. 40200 IE lip type oil seal. This cover also incorporates two lugs to carry the clutch withdrawal lever. At the rear, another cover retains the tail end bearings, houses a Paulstra No. 19800 IIL oil seal, and carries the two rear mountings of the engine and gearbox assembly.

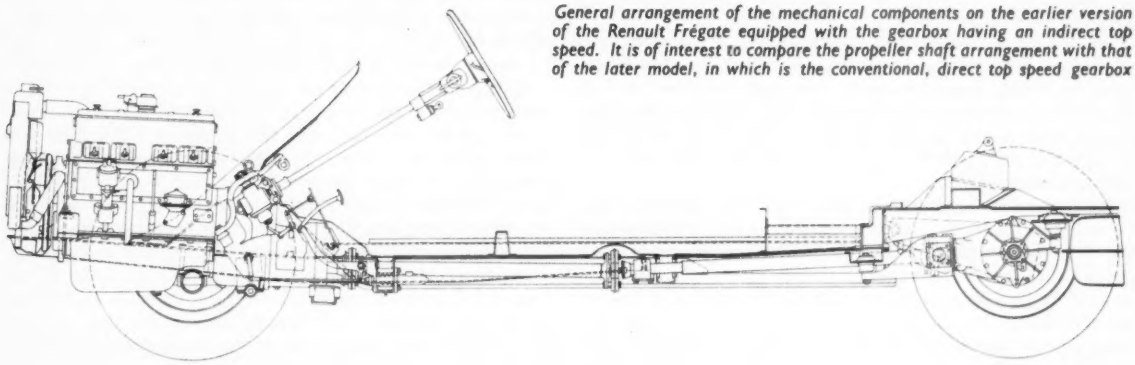
The only other cover is that on the left-hand side, which houses the selector rods. Both the front and rear covers are of aluminium alloy, while that on the side is of cast iron; their joint faces are sealed with Pâte Lowac or Perfect Seal fluid. The fact that the whole of the gear and selector mechanism can be made accessible for servicing simply by removing the one cover at the side is a good feature of this unit.

The primary shaft, which is of 38C4h specification of the French Standard PN R 971-01 (BNA448), extends forwards in the usual manner to carry the centre plate of the clutch. Its front end is 17 mm diameter where it spigots into the ball bearing in the tail end of the crankshaft and flywheel assembly. The splines that carry the hub of the centre plate of the clutch are 25 mm overall diameter and their root diameter is 21 mm. Behind the splines the shaft is 20 mm diameter, where it passes through the withdrawal bearing, and it is increased to 30 mm diameter where the lip type oil seal bears on it. A roller bearing, 35 mm inside diameter × 80 mm outside diameter, is retained between a circlip in a groove round the shaft and the primary gear, which has 19 teeth.

Formed on the rear end of the gear are the synchronizer cone, the mean diameter of which is 49 mm, and the toothed ring of the synchronizer clutch. This end of the gear is counterbored 26·2 mm to receive the needle roller bearings that carry the 20·2 mm spigot end of the mainshaft. A radial hole is drilled to join an extension of the counterbore, and breaks out into the space between the front seal and the ball bearing. Another hole is drilled from the root of the primary gear teeth into the housing for the needle rollers. Oil forced from between the primary gear teeth, through the radial hole, into the needle roller bearing, can pass out in two directions: it can come rearwards through the synchronizer assembly, or forwards and out through the radial hole drilled from the counterbore to the space between the seal and bearing. A hole is cored in the front wall of the box above the main ball bearing and oil splashed from the gears can pass through this hole into the space between the front cover and the bearings. Some of this oil also finds its way through the radial hole in the shaft into the counterbore that houses the needle roller bearings; the remainder passes through the two ball bearings that carry the primary shaft and the front end of the layshaft.

A Borg Warner type synchromesh arrangement has been adopted. The 35 teeth of the reverse gear are machined on the periphery of the internally-splined, 38C4h chrome manganese steel sleeve of the synchro unit for top and third speeds. Both synchro units are of conventional design, and are similar to those employed on a number of other vehicles, among which is the Humber Super Snipe, described fully in the July 1953 issue of *Automobile Engineer*. The central members of the synchronizers are of the same material as the sleeves, but their heat treatment is different; the synchro cups are of a material containing 57-60 per cent Cu; 0·35 per cent, max, Pb; 0·6-0·9 per cent Si; 2-4 per cent Mn; 1·5-2·0 per cent Al; 0·25 per cent, max, Fe; and the remainder is made up by Sn.

The synchronizers and gears on the mainshaft are retained between a circlip in a groove round the front end of its splined portion and a 16NC6h thrust washer interposed between the first speed gear and the inner race of the rear ball bearing. This thrust washer, together with the inner race of the ball bearing, the speedometer gear and the boss of the flexible joint, is pulled against a shoulder on the shaft by a nut and thick plain washer on the 18 mm diameter threaded tail-end. The portion of the shaft that carries the speedometer gear, and which is splined to receive the flanged boss of the flexible coupling, is 25 mm diameter. The rear ball bearing is 30 mm inside diameter × 72 mm outside



General arrangement of the mechanical components on the earlier version of the Renault Frégate equipped with the gearbox having an indirect top speed. It is of interest to compare the propeller shaft arrangement with that of the later model, in which is the conventional, direct top speed gearbox

Two views of the gearbox, showing the arrangement of the controls and of the rear mounting for the engine and gearbox unit

diameter, while the portion of the shaft that carries the gears and synchronizers is 35 mm diameter.

Only the first speed gear, which has 35 teeth, is carried on needle roller bearings, which run directly in its 41·1 mm diameter bore. The second speed gear, with 28 teeth, and the third speed gear, with 23 teeth, rotate directly on the crowns of the splines. Chamfers are machined on each spline to form a lead-in to help the lubricant to pass to the bearing surfaces between the crowns of the splines and the bores of the gears. This arrangement apparently is perfectly satisfactory since, when the gears are rotating on the shaft they are, of course, not under load. All the gears are of 36CD4h, except the primary gear, which is of 34CD4h; both these are French Standard specifications for chrome molybdenum steel.

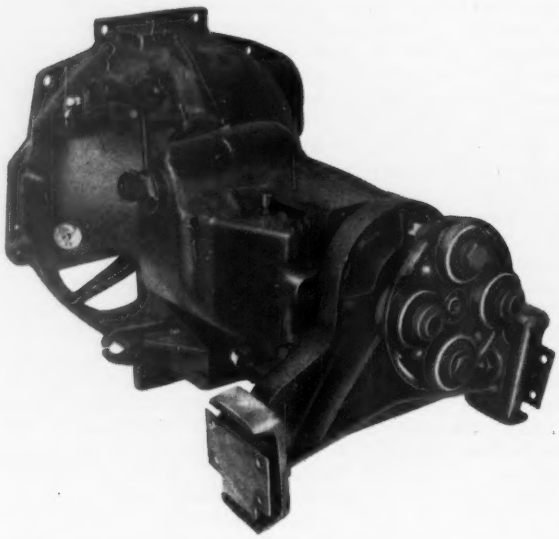
Distance washers of 16NC6h nickel chrome steel are interposed between the second and third speed gears, and between the first speed gear and the centre component of the synchronizer for first and second speeds. These distance washers are internally splined to register with the splines on the shaft. They are assembled from the front end and slid towards the rear until they reach grooves machined round the shaft to receive them. Then they are rotated one-sixteenth of a turn, so that their splines are offset relative to those on the shaft. Finally, Woodruff keys are fitted to lock them in this position.

The two ball bearings that carry the mainshaft and primary shaft assembly are located in the following manner. A groove is machined round the periphery of the outer race of each to receive a snap ring. The width of the groove is greater than the thickness of the snap ring. This is because the outer race is pulled by the end cover of the gearbox against the snap ring, which in turn is pulled against the end wall of the box. Thus, it is necessary to grind to close tolerances only one side of the groove.

As can be seen from the accompanying illustration, the oil seals at the front and rear are of a form not commonly used by British manufacturers. Although the front seal is of a fairly conventional lip type, it does not have a spring ring to hold the lip against the shaft. Moreover, its section is such that the portion of the lip that bears on the shaft has a 90 deg knife-edge.

The rear seal is entirely different. It has a spring ring moulded in the knife-edge portion that bears on the boss of the flexible-joint flange. This knife-edge portion is formed between the arms of an L-section, one arm of the L fitting against the periphery of the housing and the other arm against the rear wall of the housing. The end of this latter arm also bears on the boss of the flexible-joint flange. Evidently, the function of this arm is to prevent foreign matter from entering, while the knife-edge portion of the seal stops the oil from passing out.

It would appear that the advantage of the knife-edge type of seal is that the pressures between the seal and the rotating component are great because the bearing area is so small; this presumably gives more efficient sealing, while at the same time, since a large area of rubber is not in contact with the shaft, the heat generated locally is not too great, because it can be readily conducted away through the metal.



A one-piece forging, of 36CD4h, is employed for the layshaft and its integral gears. The constant-mesh pinion has 31 teeth, the reverse pinion 17 teeth, while the pinions for third, second and first gears have 27, 22 and 15 teeth respectively. The shaft is turned down to 20 mm diameter at its ends, which are carried in ball bearings retained between the end covers of the gearbox. Between the gears, the shaft diameter is 35 mm. For the sake of compactness, the axis of the layshaft is not directly below that of the mainshaft, but is offset approximately 18 mm to the right-hand side of the vertical plane in which the mainshaft axis lies. The centre-to-centre spacing of the two shafts is 69·99 mm. This arrangement enables the reverse idler to be positioned with its axis only slightly more than 1 mm lower than that of the layshaft and about 28 mm inboard of the plane containing the axes of the selector rods.

The reverse idler spindle is 16 mm diameter and is of 16NC6h. One end is carried in a hole in the front wall of the gearbox and the other in a boss formed about mid-way between the ends of the box. Location is effected by a dowel-ended screw passed radially through the boss and a hole in the spindle. A 24 mm long, phosphor bronze bush is pressed into the boss of the idler gear, which has 28 teeth.

All three selector rods are carried horizontally in a single vertical plane in the cover on the left-hand side of the box. Their forward ends are supported in bosses at the front end of the cover and there are three more bosses to support them about mid-way between the ends of the cover. These latter bosses also house the spring-loaded-ball type locks and the plunger type interlocks. An unusual feature of the interlock mechanism is that the periphery of the plunger in the centre rod is slotted parallel to its axis, to receive the end of a small peg pressed into a hole at right angles to that which houses the plunger. The function of this peg is to prevent the plunger from falling out of the hole when the rod is

moved to either of the two positions on each side of neutral.

The top and bottom rods terminate a short distance behind these bosses, since their forks all extend into the forward part of the box. However, the centre rod is longer, terminating at the rear of the cover, where there is another boss to support it. The selector fork on this rod, which is for first and second speeds, extends into the rear portion of the box, while the striker forks, of course, are adjacent to those on the other rods in the forward portion of the cover. Some economy has been effected by this arrangement, since the two shorter rods are only 134 mm, about 5¼ in long, while the centre rod alone is the full length of about 188 mm, approximately 7⅜ in.

The striker lever for the forward speeds is splined on to a 20 mm diameter, 16NC6h, transverse shaft carried in bronze bearings in bosses near the top of the cover. Another striker lever, for the reverse gear, is spigoted on to the boss of the first lever, to which it is welded. This twin-lever assembly can be slid axially on the splined shaft to engage the appropriate striker forks on the selector rods. Selection is effected by rotation of the splined cross shaft and striker assembly, by means of a lever retained, by a nut, against a flange on the outer end of the shaft. Another lever is splined on to the other end of the shaft and is also retained by a nut. A detent is formed in the end of this lever and a spring-loaded ball registers in it to ensure that the strikers are accurately centralized relative to their forks when they are in the neutral position. The spring-loaded detent is carried in a thimble housing, screwed into the top of the gearbox. This extra detent is incorporated because the connection between the control on the steering column and the lever assembly is of necessity relatively flexible, and although capable of transmitting the control motion it is not suitable for forcing the strikers into misaligned forks.

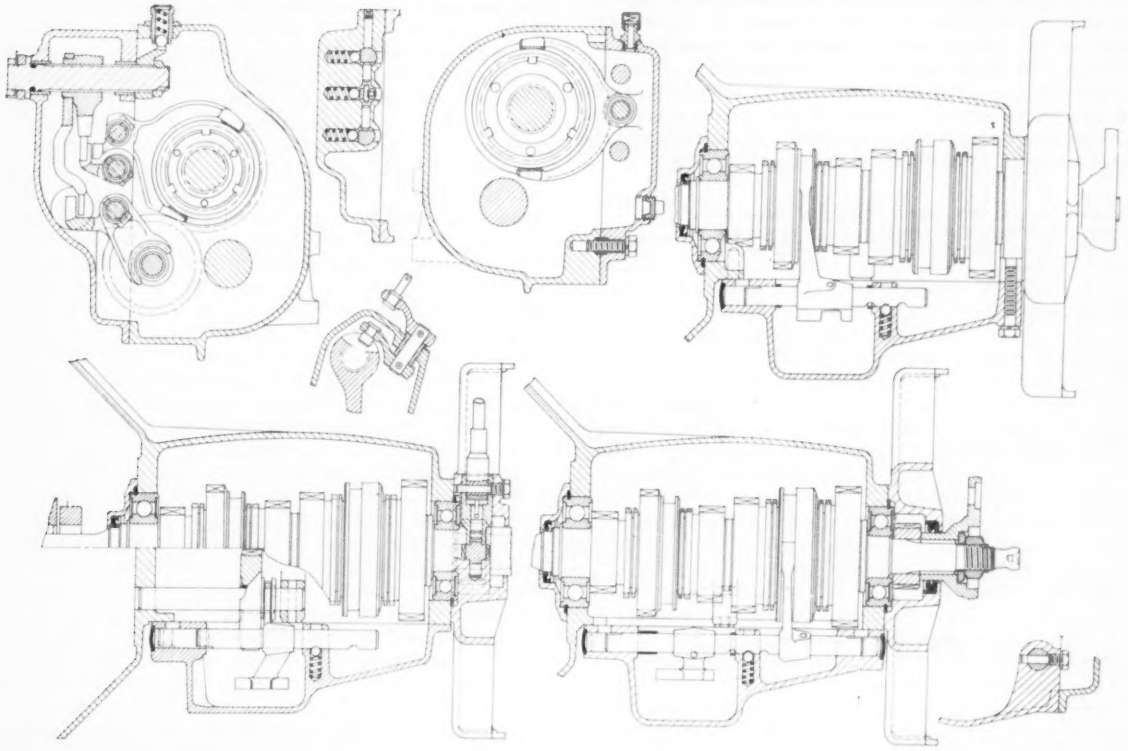
Movement of the strikers along the transverse splined shaft is effected by two levers pinned on the ends of a short 10 mm diameter spindle. This spindle is carried in a phosphor bronze bush in a boss in the casting, immediately above the strikers. One of the levers is inside the box; on its end is riveted a hardened steel peg, which registers in a slot in the boss of the striker assembly. The other lever is outside the box, and riveted in a hole at its end, is a pin for the connection to the steering column control.

A fairly complex system of levers and links connects the steering column control to the gearbox. The control lever near the upper end of the steering column is pivoted in a small fabricated turret welded radially to the main control shaft, the axis of which is set at an angle of 3 deg 7 min to that of the steering control column. This shaft is of tubular form and its upper end spigots on to a rod carried in a bracket clamped to the top of the steering column. A hole is drilled diametrically through this rod; it is chamfered and receives the spherical end of the control lever. This lever is carried, on a tubular pivot pin, between the lugs of a small steel casting spigoted into the end of the turret. The pin is split longitudinally and, before assembly, is larger than the holes in which it fits. This, of course, is an anti-rattle measure. Nylon washers are interposed between the lugs and the lever not only to obviate rattles, but also to reduce the rate of wear. A small cap, of approximately hemispherical form, is fitted over the whole assembly to protect it from foreign matter and to give it a finished appearance.

From this description and the illustration, it can be seen that when the lever is moved up and down, the turret and control shaft assembly rotates about the axis of the shaft, and when the lever is moved backwards and forwards, the assembly is moved axially. A nylon bush is housed in a counterbore in the upper end of the shaft and retained by a wire circlip. Lubrication is not required at any of the bearings or pivots at this end of the assembly; in fact, only one of the other pivots in the control linkage is lubricated.

Since the steering box is mounted on the outer face of

A noteworthy feature of the Frégate gearbox is that all the mechanism is accessible when the side cover, carrying the selector rods, is removed



one of the two bracing struts, between the dash and the main longitudinal members that extend forward to carry the engine and suspension, it is a fairly large distance from the longitudinal axis of the chassis. This distance, in fact, is about 367 mm, and the lower end of the gear shift control is about 68 mm inboard of the centre of the flexible coupling that connects the steering column to the steering gear. A 12 mm diameter rod is riveted in the lower end of the tubular, gear-control shaft, which is carried by a spherical bearing in an aluminium, pressure die-casting bolted to the lower end of the dash panel, adjacent to its junction with the toeboard. An interesting feature of this bearing is that both the spherical portion and the two cup components are of nylon. It is assembled into the rear end of the casting and retained by a circlip.

To apply the load necessary to take up the clearance between the cup and spherical components, a spring is fitted in a counterbore forward of the spherical bearing. A steel washer is interposed between the end of this spring and the adjacent cup component. The forward end of the spring bears on a spherical washer, of Superpolyamide A.P, which closes the forward end of the casting that houses the assembly. Two mild steel washers, one each side of the Superpolyamide one, prevent foreign matter from entering the assembly.

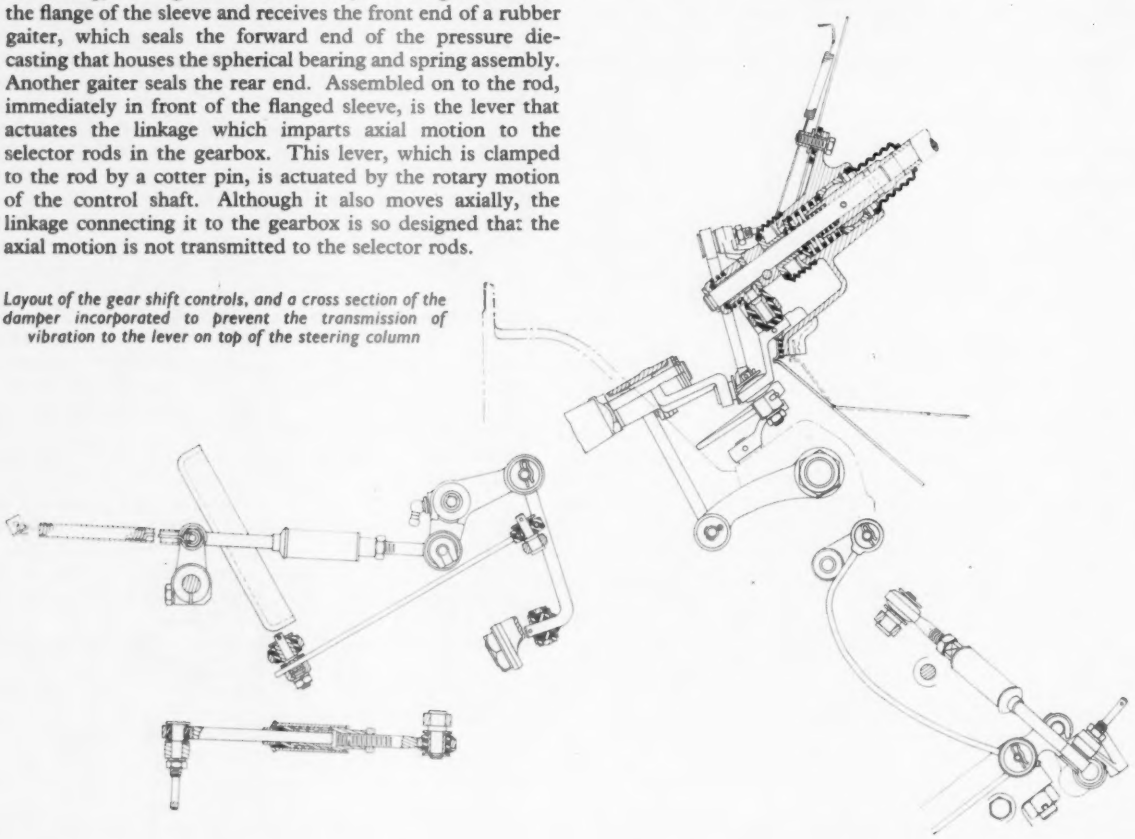
Another coil spring, which is arranged coaxially with the first, is interposed between the end of the tubular portion of the control shaft and this spherical washer assembly. The function of this spring is to ensure that the seal is fully effective and to spring-load the control in such a way that the hand lever tends to remain in the position for selection of either of the top two gears.

Assembled on to the shaft, immediately in front of the spherical washer, is a flanged sleeve made of Superpolyamide D. A ring, of flanged channel section, fits in a groove round the flange of the sleeve and receives the front end of a rubber gaiter, which seals the forward end of the pressure die-casting that houses the spherical bearing and spring assembly. Another gaiter seals the rear end. Assembled on to the rod, immediately in front of the flanged sleeve, is the lever that actuates the linkage which imparts axial motion to the selector rods in the gearbox. This lever, which is clamped to the rod by a cotter pin, is actuated by the rotary motion of the control shaft. Although it also moves axially, the linkage connecting it to the gearbox is so designed that the axial motion is not transmitted to the selector rods.

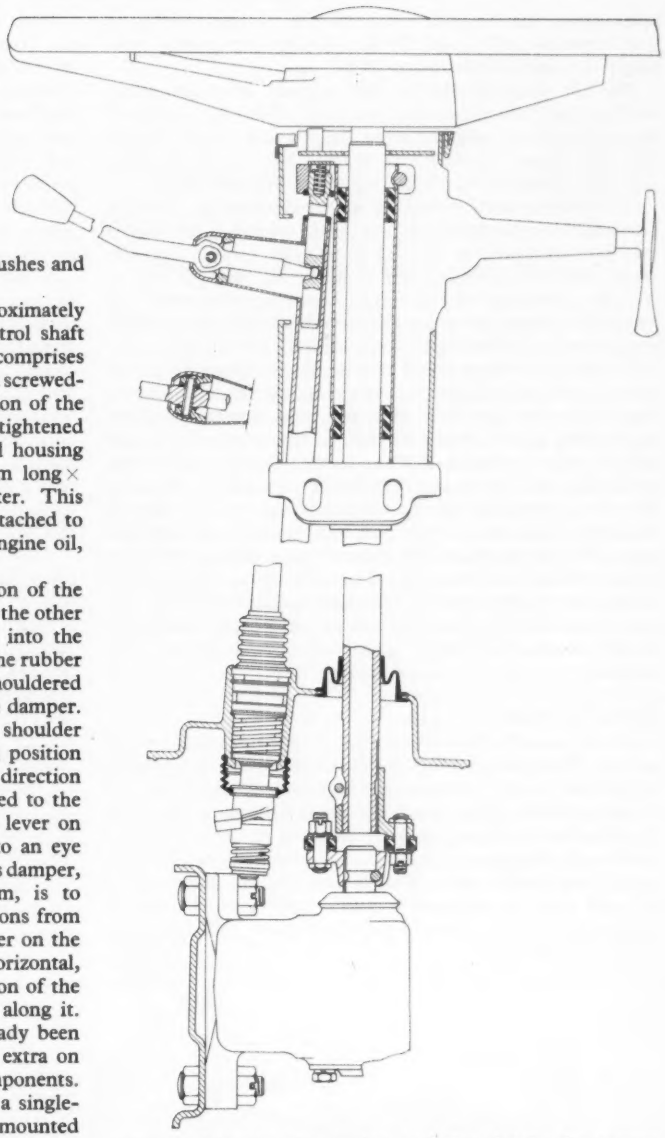
Axial motion of the control shaft is transmitted by a flanged collar with a peg pressed into a radial hole in the flange. A steel washer is interposed between this collar and the lever secured to the shaft by the cotter pin, and the whole assembly is retained by a coil spring, a dished washer and a circlip on the end of the rod. The coil spring fits round the collar and is compressed between the flange and the dished washer. A rubber sealing ring, of circular section, fits in the dished portion of the washer and bears against the end of the collar to seal the two bearings that carry the flanged collar. These bearings are sintered bronze bushes, of the self-lubricating type. Each is 10 mm long and the space between them is approximately 2 mm.

The peg that projects from the radial hole in the flange registers in a Superpolyamide D bush, housed in a Neoprene 60 thimble in an eye at one end of a bell crank lever. This bell crank lever is pivoted in the pressure die-casting that houses the spherical bearing at the lower end of the control shaft. A bowed rod connects its other end to the lever on the gearbox that moves the striker transversely, for the engagement of one or other of the striker forks. The conical faced, circular eye-fitting at one end of the rod and the flanged bush at the other end are of hard rubber, so that lubrication is not required at these points.

Transmission of the rotary motion of the control shaft is effected by a slightly more complex linkage. The lever secured by a cotter pin to the shaft on the steering column is connected by a tie rod, which incorporates a damper, to a bell crank lever pivoted on the clutch housing, whence another tie rod transmits the motion down to the operating lever on the side of the gearbox. The bell crank lever on the clutch housing is carried on two flanged nylon bushes on a 14 mm diameter pin. A grease nipple is provided for the lubrication of these bearings. All the pivots at the ends of



Layout of the gear shift controls, and a cross section of the damper incorporated to prevent the transmission of vibration to the lever on top of the steering column



Steering column and gear shift control. The gear shift lever is carried on a split tubular pivot, which is sprung into its hole, so that there is no clearance, and therefore no rattling. Nylon washers, interposed between the lugs and the eye of the lever, also help to prevent rattles from developing as a result of wear after extensive service

the tie rods are equipped with fairly hard rubber bushes and therefore do not require lubrication.

Of particular interest is the damper in the approximately horizontal link between the lower end of the control shaft and the bell crank lever on the clutch housing. It comprises a cylindrical housing, one end of which is closed by a screwed-in adaptor to receive the threaded end of the portion of the tie rod attached to the bell crank lever. A lock nut is tightened against this end. The other end of the cylindrical housing is lipped internally to retain a rubber bush, 9 mm long × 7½ mm inside diameter and 14 mm outside diameter. This bush acts as a seal round the portion of the rod attached to the lever on the control shaft, and it retains the engine oil, which is introduced to lubricate the unit.

There are two steel sleeves pinned to this portion of the rod, one immediately in front of the rubber seal and the other at the extreme end of the rod, where it projects into the damper. These sleeves are 26 mm apart. Between the rubber bush and the sleeve on the end of the rod is a shouldered sleeve that limits the axial motion of the rod in the damper. A light coil spring, in compression between the shoulder and the rubber bush, retains the rod at the mean position of its travel in the damper. Pulling in the opposite direction is a tension spring; one end of this spring is secured to the pin by means of which the rod is attached to the lever on the control shaft, and the other end is hooked into an eye in a lug on the vehicle structure. The function of this damper, which has a total stroke of approximately 2 mm, is to prevent the transmission of small amplitude vibrations from the engine and gearbox assembly to the control lever on the steering column. Since the rod is approximately horizontal, the large amplitude vibrations, due to vertical motion of the engine and gearbox assembly, are not transmitted along it.

The Transfluide transmission which, as has already been mentioned, is to be offered shortly as an optional extra on the Grand Pavois model, comprises well proved components. From the engine, the drive is transmitted through a single-stage torque converter, the reaction member being mounted on a free-wheel on the output shaft. This torque converter is supplied with oil from the engine lubrication system. The oil passes through a filter mounted on the engine and thence is piped to the rear of the converter, where it enters the chamber at a point between the impeller and reaction members. Oil leaves the converter through a hole drilled axially from the forward end of the output shaft, in which is incorporated a non-return valve; it is, of course, discharged into the engine sump. The maximum torque multiplication ratio obtained from the converter is 2·3 : 1.

From the converter, the drive is taken through an electro-magnetic clutch of the single-dry-plate type. This clutch is disengaged only when the gear shift control lever is moved to effect changes in gear ratio. The gear lever incorporates an electrical switch, which is normally open, but is closed when the lever is being moved. Closure of the switch completes a circuit to disengage the clutch magnetically.

Behind the clutch is a conventional gearbox, giving three forward ratios and one reverse. Single-helical, constant mesh pinions and gears are employed throughout, and each of the three forward gears is equipped with a synchromesh mechanism. Because of the interposition of the hydraulic converter between the engine and the clutch, the torque transmitted through the box may be more than double the engine torque. Therefore, the gear assembly, as well as the clutch, are of heavier proportions than would be required in a more conventional transmission.

It is not intended that frequent gear changes should be made: the top, or direct speed, is designed for motoring on the open road, second speed is for driving in mountains and overtaking on hills, while the first gear is designed to meet special requirements, such as for restarting on steep hills, slow-speed manoeuvring, and possibly engine braking. In each gear, the converter gives torque multiplication, within its range, according to the load; it also provides for engine braking in the over-run condition.

For starting from rest, top gear is normally engaged. Therefore, the overall ratio initially is that of the rear axle, 3·89 : 1, multiplied by the maximum ratio of the converter, which is 2·3 : 1. This gives an overall ratio of 8·95 : 1. As a steady speed condition is approached, the torque converter ratio changes progressively towards unity. It is claimed that when the converter is fully coupled, the transmission efficiency is 97 per cent. The overall ratios in the

other gears are: second, between 6·30 : 1 and 14·49 : 1; first, between 8·87 : 1 and 20·40 : 1; while the reverse gear range is between 9·80 : 1 and 22·54 : 1.

The six positions for the gear selector lever are: park, reverse, first, neutral, second, and top. They are indicated respectively on a quadrant above the steering column by the following letters: P, R, E, N, M, and VR. In the parking position, a transmission lock is brought into operation.

To offset to some extent the losses inherent in a torque converter type transmission, the power output of the engine has been increased to 80·3 b.h.p. at 4,000 r.p.m. This has been done by increasing the compression ratio to 7·5 : 1. To take advantage of the torque converter characteristics, the engine torque curve has also been modified, now peaking at 2,500 r.p.m. The maximum torque is 124·3 lb-ft.

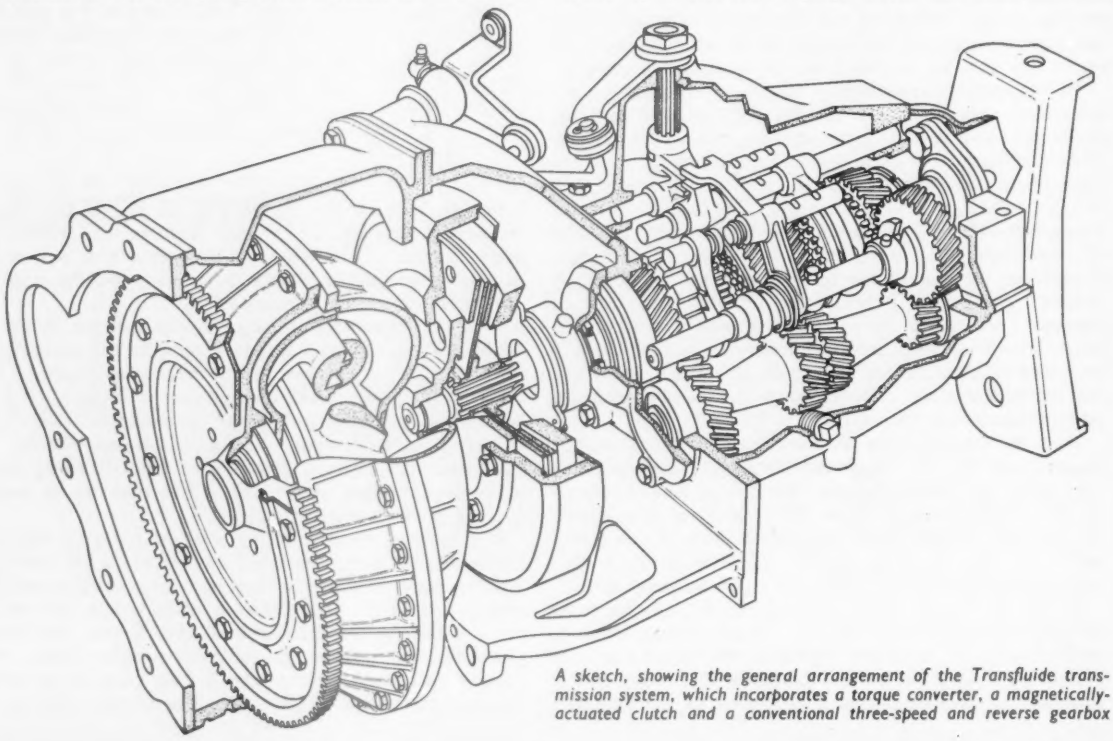
There can be little doubt that an electro-magnetic clutch offers a relatively simple solution to the problem of designing semi-automatic and fully automatic transmissions without appreciable loss in overall efficiency. However, many motor manufacturers consider that the friction facings in this type of clutch, used alone, can hardly be expected to be satisfactory in automatic transmissions, in view of the slip that would be inevitable during gear changes and take-up from rest. The employment by Renault of a torque converter obviates difficulties due to overheating and wear of the clutch facings; it smooths out gear changes and take-up from rest; and, because of the range of torque conversion obtained, it enables a relatively simple gear shift control system to be adopted.

Propeller shaft

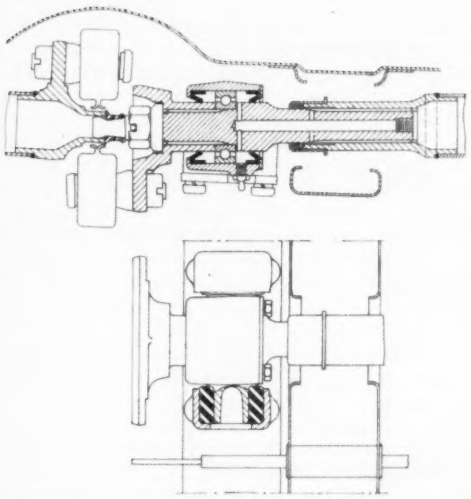
As has already been stated, the propeller shaft is in two pieces. The front portion is 875 mm long and the other is 1,075 mm long, both measured between the centres of the flexible joints. They are each 45 mm outside diameter. Layrub flexible joints are employed and all are of identical design. Each comprises a pair of two-armed spiders, between which is a flexible centre-plate assembly. One of the spiders of each joint incorporates a central spigot, which registers in a socket on the other, to provide accurate radial location. A thick rubber sleeve is fitted over the stem of the spigot and bears against the end of the socket to seal the assembly against the entry of abrasive matter. A ball end, 17½ mm diameter, is formed on the spigot to allow for slight angular misalignment. The centre-plate assembly comprises two steel discs with two cup-shaped depressions pressed into each, and they are assembled together with their cupped portions opposite one another to form the housings for the rubber bushes, which are the flexible elements of the drive. There is a flanged hole in the centre of each plate to clear the spigot and socket assembly.

A sliding joint is incorporated at the front end of the rear portion of the propeller shaft. The effective length of engagement of the splines is about 55 mm and the overall diameter of the splined portion of the centre component is 25 mm. Two rectangular section seals are housed in a pressed steel cap assembled over the end of the outer component. The rearmost seal is of rubber to retain the lubricant, while the forward seal is of felt to keep out abrasive matter. Both seals, of course, bear on the shank on the centre component of the sliding joint, just forward of the splined portion. The front end of this shank is splined to receive the flexible joint in front of the intermediate bearing.

This intermediate bearing assembly comprises a single roller bearing in a cylindrical housing, the front and rear ends of which are closed by lip-type oil seals. The front seal is assembled into the cylindrical housing from the rear and is retained by an inwardly-turned lip at the forward end of the housing. Behind it, the outer race of the ball bearing is free to float axially in the housing. The rear oil seal, which is assembled last of all, is retained by a plate bolted to the end face of the housing. The inner race of the ball bearing is clamped between the boss of the flexible joint and a shoulder on the shank. A grease nipple is screwed into the base of the housing, in such a position that the lubricant is injected into the space between the rear seal and the ball bearing. Thence it flows not only into the ball bearing but also through a hole drilled diametrically through the shank and into an axial hole drilled from the rear end of the shank, where it is sealed by a screwed-in plug. From this axial hole, the



A sketch, showing the general arrangement of the Transfluide transmission system, which incorporates a torque converter, a magnetically-actuated clutch and a conventional three-speed and reverse gearbox



lubricant passes through another diametrically-drilled hole to the splines of the sliding joint.

Considerable pains have been taken to prevent the transmission of vibration from the intermediate bearing to the vehicle structure. Two lugs extend laterally from each side of the cylindrical housing. The pair on one side are horizontal, while that on the other side are vertical. Each pair projects between the arms of a channel section bracket, and rectangular section rubbers are interposed between the lugs and the arms of the brackets. This assembly not only damps out high frequency vibrations, but also enables the bearing housing to align itself automatically relative to the axis of the drive shaft.

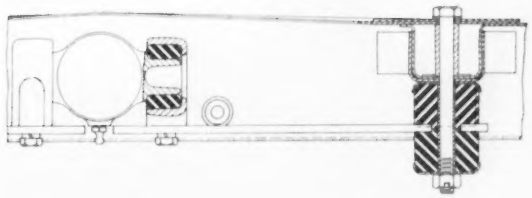
Both brackets are bolted on top of a single-leaf spring, mounted transversely between the main longitudinal members of the vehicle underfloor structure. This spring is of 45S8e, a steel listed in the French Standard NF PN R 971-01 (BNA 448). It is 6 mm thick × 20 mm wide, and the distance between the centres of its supports is 500 mm. The function of this spring is to absorb vertical vibratory forces.

Each end support comprises a bolt assembled vertically through the longitudinal member of the underfloor structure and two circular section rubbers, one above and the other below the spring end, a tubular distance piece and two dished retainer washers, one above and the other below the rubbers. A nut is tightened against the lower retainer washer to compress the rubbers, the amount of compression being fixed by the distance tube round the shank of the bolt. The hole in the leaf spring, through which the bolt passes, is of large diameter and the rubbers are spigoted into it to prevent metal-to-metal contact between the tube and the spring.

Final drive

On the earlier versions of the Frégate, a large cover on one side of the spiral-bevel differential casing housed one of the two bearings that carried the differential cage, while the other was in a housing in the casing. The latest models, however, which have a hypoid type final drive unit with a pinion offset of 28·575 mm, have two small covers, one on each side, to retain the bearings in their housings in the casing; also, there is a large cover bolted to the rear of the unit to give access to the gears. This represents an advance in design, because the whole gear assembly is now carried by a one-piece rigid casting. Moreover, the centres of the taper rollers of the bearings are now 173 mm instead of 153 mm apart. This wide spacing of the bearings has enabled stiffer differential gears to be employed and a one-piece instead of a two-piece cage to be used.

At the front end of the unit, the 20NC6h nickel chrome



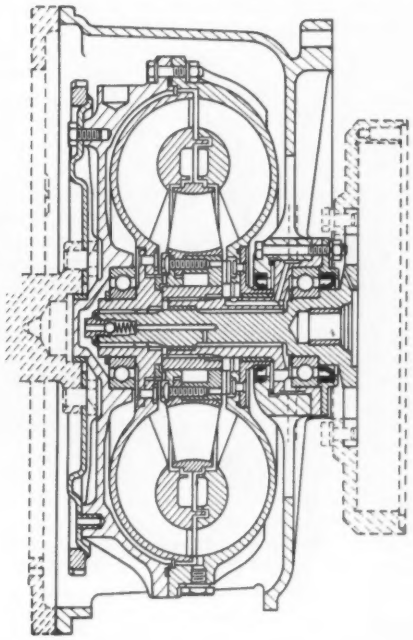
Vibration-isolation measures for the propeller shaft bearing

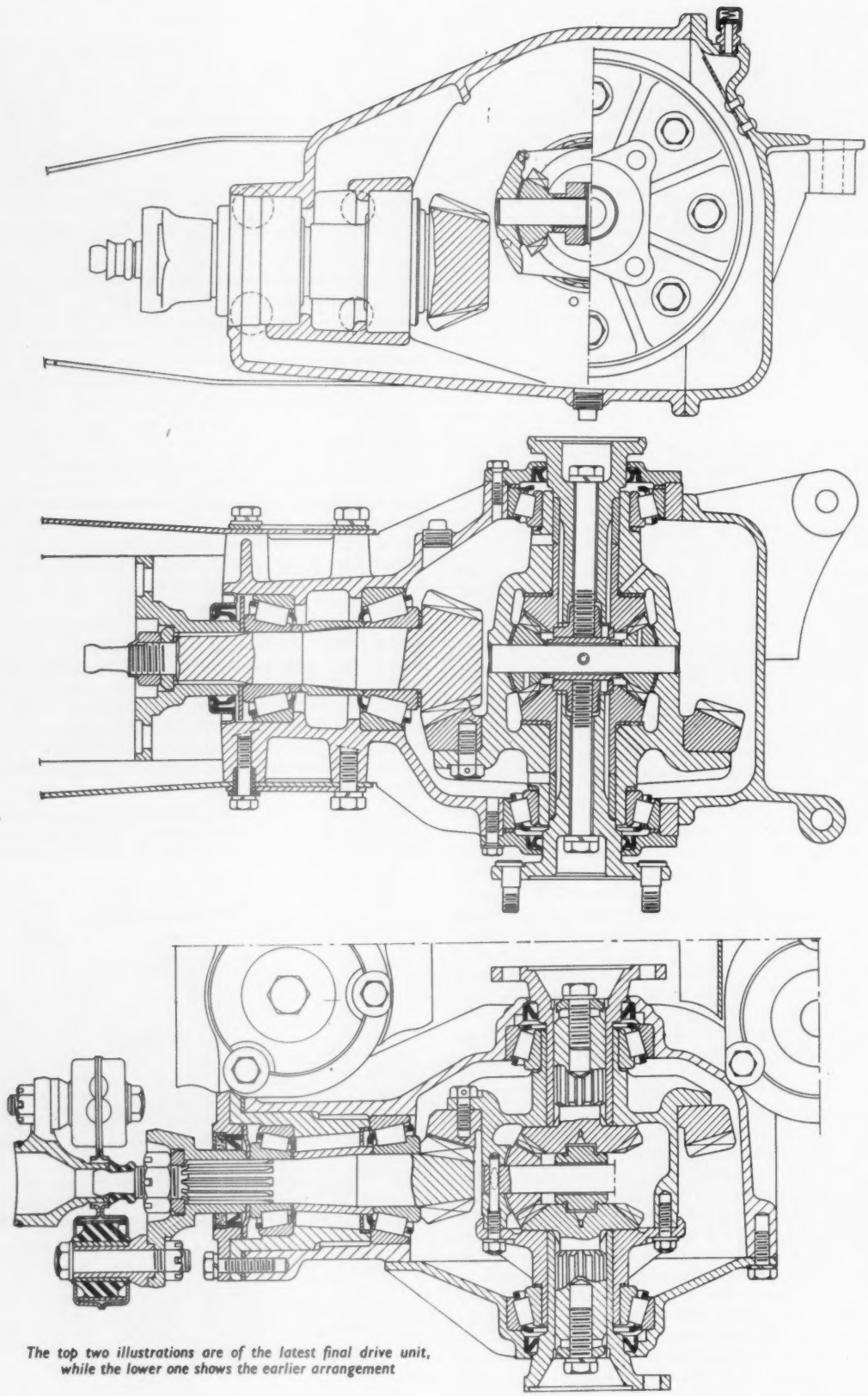
steel hypoid pinion shaft is extended to form a ball-ended spigot to centralize the flexible coupling. Immediately behind this spigot, the shaft is threaded 18 mm diameter for the nut that retains the coupling flange for the universal joint, and the bearing assembly. This nut has an integral sleeve extension, which is peaned locally into a longitudinal slot in the threaded portion of the shaft, to lock it after assembly. The nut is pulled against a thick washer, which in turn bears against the boss of the flange splined on to the 30 mm diameter portion of the hypoid pinion shaft.

A lip-type oil seal, housed in the front end of the cast iron differential casing bears on the periphery of this boss. Like the two seals on each side of the differential gear assembly, it is unusual in that it is moulded with two lips bearing on the boss, one facing outwards to keep out water and abrasive matter and the other facing inwards to retain the lubricant. These lips are so designed that they make only line-contact round the boss, so that the pressure between the sealing faces is higher, while at the same time friction is lower and there is less heat to be conducted away. Interposed between the boss of the flexible joint and the inner race of the adjacent taper roller bearing is a large diameter thrower disc, which prevents oil from being flung in large quantities on to the seal. This is necessary because of the slope of the unit.

The two taper roller bearings that carry the hypoid pinion

Engine oil is employed in the Transfluide torque converter, and the inlet and outlet ducts, which are shown dotted in this illustration, are respectively above and below the rear ball bearing, which is between two seals





The top two illustrations are of the latest final drive unit, while the lower one shows the earlier arrangement

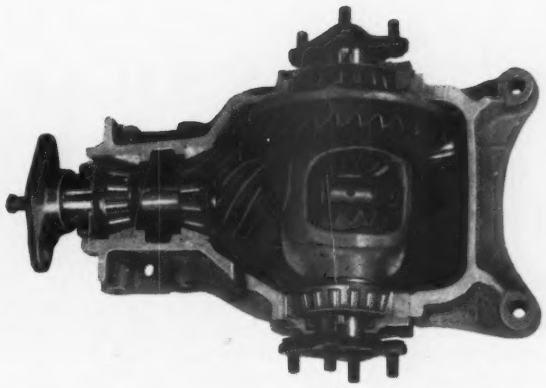
are clamped between the thrower disc and the pinion. They are separated by a tubular distance piece and a washer used to adjust the preload on the bearings. This preload is such that the torque required to turn the shaft, with neither the oil seal nor the crown wheel in position, is 0·180-0·280 m-kg. The axial position of the pinion is regulated by another distance washer between it and the inner race of the rear bearing. Both outer races abut against shoulders in the nose piece of the casting. The centres of the rollers are approximately 66 mm apart. Their dimensions are: front bearing, inner race 30 mm inside diameter × 28·75 mm long; outer race 72 mm diameter: rear bearing, inner race 35 mm inside diameter × 32·75 mm long; outer race 80 mm diameter.

The 186 mm diameter, 16NC6h, nickel chrome steel crown wheel is spigoted on to the flanged one-piece differential cage in the usual manner. It is retained by eight 10 mm diameter set bolts. Two taper roller bearings carry the cage. Their inner races are 50 mm inside diameter × 21·75 mm long and their outer races are 90 mm outside diameter. The preload on these bearings is adjusted by shims between the inner races and shoulders on the cage, while the meshing of the crown wheel and pinion is regulated by shims between the outer races and the plates that are bolted to the outer faces of the casing to retain the bearings. These plates also house the double-lip seals that bear on the bosses of the flanges for the universal joints at the inner end of the swinging half shafts.

There are a number of unconventional features in the differential pinion and gear assembly. A 16 mm diameter, 16NC6h differential pinion spindle is employed. Assembled on it are two phosphor bronze spherical thrust washers, and the two 16NC6h differential pinions of about 50 mm diameter, with an XC10f mild steel distance sleeve between them. The whole assembly is retained by a tube in a 6 mm diameter hole drilled diametrically through the centre of the pin. This tube is split along the whole of its length and in the free condition is larger in diameter than the hole into which it is sprung. Its ends project into larger diameter holes in the distance sleeve. Flats are machined diametrically opposite one another on the periphery of the distance sleeve. These flats are parallel to the axis of the split retainer tube. They form abutment faces for the inner ends of the differential gear assemblies.

The differential gears are of 16NC6h and are 74 mm overall diameter. Each is splined on to the flange for the universal joint at the inner end of the swinging half shaft, and is retained by a special nut and a 12 mm diameter bolt passed from the outer end, through an axial hole in the boss of the flange. The nut, which is also of 16NC6h, is spigoted into a counterbore in the inner end-face of the gear. It is prevented from rotating by a peg passed through a hole, near the periphery of the nut, to project between a pair of splines in the gear. The inner face of this nut abuts against the adjacent flat on the distance sleeve between the differential pinions. When the bolt and special nut are tightened, the differential gear is pulled against a distance sleeve, which abuts against a shoulder on the boss of the flange for the universal joint. An interesting feature of this assembly is that the flat thrust washer between the differential gear and the cage is of Bakelite plastics material, which is perfectly satisfactory because there is no problem with regard to heat dissipation from the bearing surfaces.

Since the location of the road wheel assembly is effected entirely by the suspension, the swinging half shafts transmit only the driving torque. Each shaft comprises a 151 mm outside diameter × 147 mm inside diameter tube, into one end of which is spigoted the Glaenser-Spicer, needle roller type, universal joint connecting it to the differential unit, and into the other end the centre component of the sliding joint. These two end fittings are welded in position. The



Final drive unit sectioned to show the gears and their bearings

sleeve component of the sliding joint is integral with one pair of arms of the universal joint. Between these arms, a domed washer seals the outer end of the sleeve of the sliding joint. A 2 mm diameter hole in the centre of the washer prevents it from being forced out by the pumping action. A seal, in a cupped housing screwed on the other end of the sleeve, bears on the inner component of the sliding joint. The splines of the inner component are about 70 mm long and a 21 mm long groove is machined round them, mid-way between their ends, to form a reservoir for the grease, which is inserted through a nipple screwed radially into the sleeve component.

Both universal joints are the same size. The rollers are 10 mm long, their outer races are 21 mm inside diameter and their inner races 17 mm outside diameter. The outer races are of thimble form, their closed ends being outermost to retain the lubricant. A small felt seal is fitted between the spider and the inner end of each of these outer races. Measured between the centres of the universal joints, the length of each swinging half shaft is about 463 mm. Its angular movement is ±7 deg from the mean position, which in the laden condition is inclined 1 deg up from horizontal. Since the axes of the differential gears and the stub axles are parallel, there is no change in velocity as the wheels rise and fall.

The general arrangement of the wheel hubs will be shown in an illustration in Part II of this article. One end of the 42CD4h, selected chrome molybdenum steel stub axle is forked to form the spider that carries the two needle roller bearings of the universal joint, while the other is splined to carry the driving flange. This flange, together with the bearing assembly in the wheel hub, is pulled on to the stub axle by a nut on the 20 mm diameter threaded outer end. The driving flange is centralized by a conical washer interposed between it and the nut.

Both the ball bearings that carry the stub axle in the hub are 31 mm inside diameter, but the outer race of the inboard bearing is 72 mm diameter, while that of the outboard bearing is 62 mm diameter. Their centres are approximately 60 mm apart. Axial location of the stub axle and bearing assembly is effected by a flanged plate, which is spigoted into the inner end of the hub to pull the outer race of the inboard bearing against a shoulder in its housing. The outer race of the outboard bearing is not positively located axially in its housing.

An oil seal with a line-contact type lip, as previously described, is fitted in each end of the hub assembly. In addition, a rubber sealing ring is interposed between the hub and the flanged spigot that retains the outer race of the inboard bearing. Should any lubricant escape past the outboard seal, it is caught in a dished steel trap secured, together with the brake drum, to the wheel driving flange.

CRANKSHAFT GRINDING

New Developments in Newall Machines and Techniques

UNTIL recently the Newall Engineering Co. Ltd., Peterborough, marketed only a range of semi-automatic, heavy duty crankpin grinders. The original range has now been supplemented by the development of two new models, both based on the model HAC machines. The first of these, designated the U/HAC, was initially designed for an American customer. It is a heavy duty universal machine with 28 in swing, and the maximum and minimum distances between the faces of the workhead and tailstock spindle flanges are 84 in and 36 in. This machine is designed for re-grinding or grinding on a production basis both main journals and crankpins. It incorporates a fully automatic grinding cycle by hydraulic and electric power.

Both workheads are driven by a common drive-shaft from a 5 h.p., 960 r.p.m. motor. This arrangement eliminates several disadvantages that may be encountered when a loose headstock drive is transmitted by the workpiece, and it also ensures accurate radial positioning of the crankpins. A seven-speed gearbox provides work speed of 35, 41, 47, 52, 58, 64, and 72 r.p.m., to cover all normal requirements for crankshaft work. The right hand workhead provides for lateral adjustment of the centre by a hand wheel control to facilitate loading during the main journal grinding operation.

Wheelhead

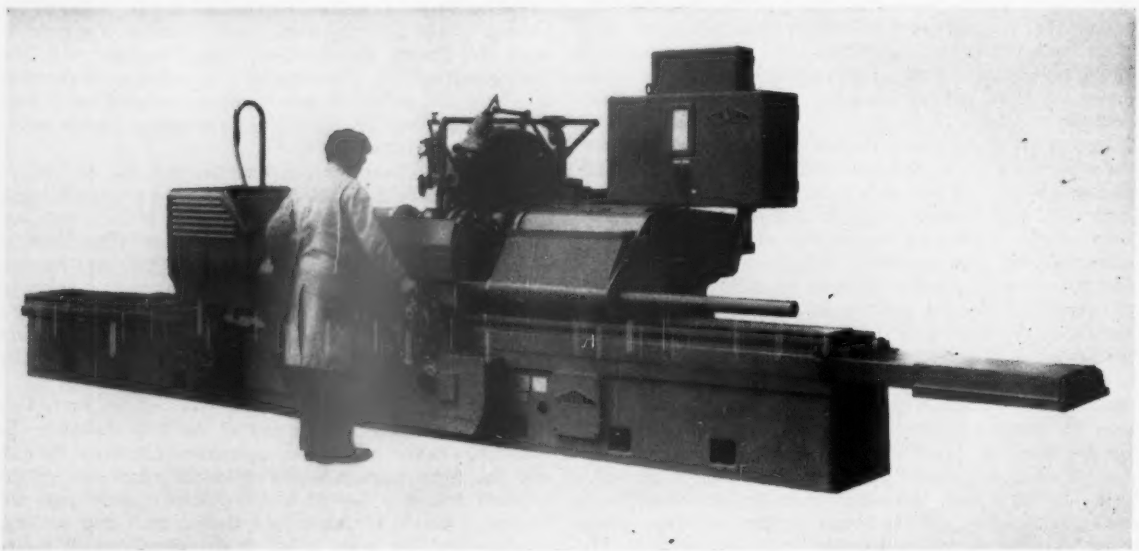
A heavy-duty spark-splitter type wheelhead is employed. It mounts a 42 in diameter wheel and has a 6 in rapid approach. Four automatic variable rates of infeed are provided. One is for face grinding and three for diameter grinding, the latter giving in effect a diminishing feed. A micro-adjusted diamond dresser may be clamped in either workhead for dressing a wheel to correct width. In addition, a table-mounted radius dressing attachment is available for journal grinding, and a dresser bar-mounted radius attachment is supplied for crankpin grinding.

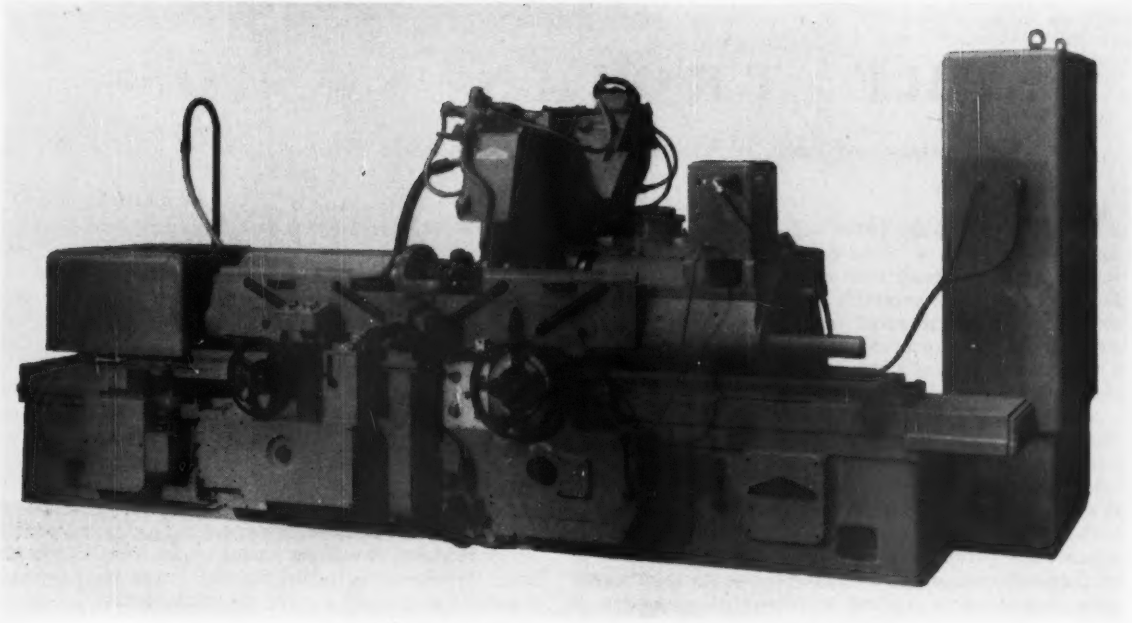
The table traverse is hydraulically operated at steplessly variable rates between approximately 3 in and 160 in by a single joy-stick control. Automatic positioning of the bearing or pin and the grinding wheel is effected by a spacer bar and hydraulic plunger located on the front of the machine. Separate spacer bars for crankpin and journal grinding are supplied as standard equipment to suit customers' requirements. Both throw blocks are easily detachable from the work spindle flanges to facilitate change over from crankpin to journal grinding. The left-hand member is designed to accommodate a dividing plate suitable for the type of crankshaft concerned. A Newall electro sizing gauge is fitted to determine finished diameter and control the automatic cycle of the wheelhead. Isolator switches are provided to allow wheelhead approach and retraction to be controlled manually.

Automatic cycle

The automatic grinding cycle is identical for both pins and journals. It has the following sequence:—The machine table is positioned by a hydraulically operated plunger locating in a spacer bar. Forward movement of the joy-stick starts the workhead and approachable wheelhead. The wheelhead has a quick approach to a predetermined position at which the infeed rate slows and the crankweb faces are ground. Following this, the infeed rate is again reduced, coolant begins to flow and the wheel begins to grind the work diameter. The Newall electro sizing gauge is applied to the workpiece and feed continues until the work is between 0·005 in and 0·001 in above the finished diameter. At this stage a yet slower or semi-finishing feed comes into operation. The final infeed rate comes into operation slightly before

The Newhall U/HAC universal heavy-duty crankshaft grinding machine





The Newall A/HAC crankpin grinding machine with special automatic cycle and gauging system

the finished size is reached, and the electro sizing gauge takes control and withdraws the wheelhead immediately the correct diameter is established.

Automatic crankpin grinder

In addition to the machine described above, the Newall Engineering Co. has recently completed the first of a new series of automatic crankpin grinders. This machine, model A/HAC, has been developed for large scale production of four- or six-cylinder automobile crankshafts. It is designed with push-button operation that initiates the following cycle:—

After the shaft is loaded and clamped in position, the start button is pressed to bring the wheelhead forward at rapid approach and to start rotation of the shaft. Rapid approach continues to a point just clear of the faces to be ground. The wheelhead then proceeds to feed in at a predetermined rate until the faces are finished. A further roughing feed is then applied automatically until a diameter within approximately 0·006 in of finished crankpin size is reached. At this point the feed is further retarded, and a steady rest comes into position, slowing its rate of approach in the final stages to permit a Sigma air sizing gauge to locate on the pin. Feed continues until a signal, indicated by a light, stops the feed and creates a dwell period. The grinding operation ceases on receipt of a further signal, transmitted when the finished diameter is reached. Automatic rapid withdrawal of the wheelhead follows, the steady rest and sizing gauge retract to their inoperative positions, and work rotation ceases.

On reaching its rear position, the wheelhead operates a control circuit, which indexes the table to a position relative to the pin in line with that already ground. On final location in the spacing bar, the table locking plunger engages a micro-switch to initiate repetition of the grinding cycle. After completion of the second crankpin and return of the wheelhead, the table returns automatically to the grinding position for the first pin in readiness for another shaft to be loaded. The automatic cycle can be arrested at any point by application of a single push-button.

Safety interlocks are provided for the following purposes: (a) to prevent the cycle being started until the workpiece is clamped; this is effected by a pressure switch incorporated in the hydraulic feed line to the clamps. The switch will not operate if for any reason hydraulic pressure fails: (b) to obviate movement of the table unless the workheads are in the correct rotational position; a cam operating a switch mounted on the work spindle performs this function. By means of a single rotary switch the entire automatic cycle can be disengaged and a set of control buttons for manual application during setting up be brought into operation simultaneously. These controls are for advancing the wheelhead, retracing the wheelhead, and for indexing the table to the requirement of the component to be ground.

In addition to the automatic features and gauging system, the machine is provided with a hydraulically operated wheelhead-mounted dresser, with three diamonds that permit both radii and front face of the wheel to be finish dressed in one operation. When this dresser is used, the feed advancing the diamond is applied by push-buttons, which simultaneously advance the wheelhead and thereby compensate for reduction in wheel size. The controls for the dresser are duplicated at the rear of the machine for use during setting up. A safety device prevents traverse of the diamond carriage unless the wheel is in the correct position for dressing. To prevent fluctuation in the hydraulically operated feed rates owing to temperature changes, a unit is included in the hydraulic tank to maintain a constant ambient temperature.

Air Suspension Systems

ONE of the outstanding features of the recent Frankfort motor show was the wide application of air suspension systems to German public service vehicles. In fact, only two of the major German manufacturers failed to show public service vehicles with air suspension, and these two are carrying out development work in this field. This is a development with great possibilities, and because of this, the December issue of this journal will include a detailed and authoritative article on the German development.

THE TROJAN SENIOR

A Diesel-engined 1-tonner Now Fitted With Independent Front-Wheel Suspension

ABOUT five years ago Trojan Ltd., Purley Way, Croydon, Surrey, introduced a diesel-engined 20-cwt commercial vehicle which has since been continuously in production. For three years no comparable vehicle of this capacity was available. It was produced in a number of versions—a standard delivery van, a long-wheelbased, large-capacity van, a personnel wagon and a pick-up vehicle—as well as in bare chassis or chassis and cab form.

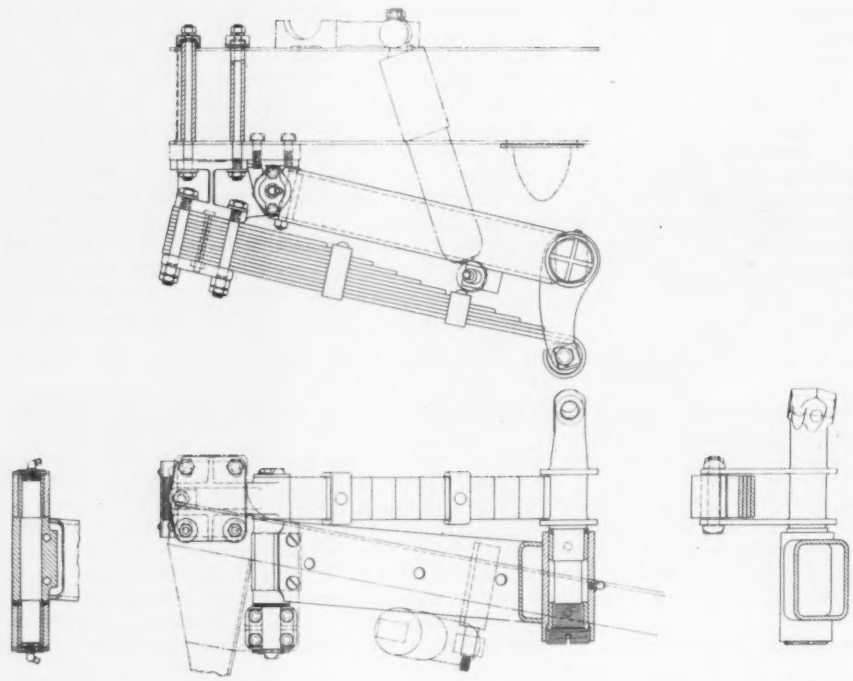
Suspension was on conventional lines with a beam-type front axle and half-elliptic springs fitted all round. Performance on the road was completely satisfactory and the vehicle was well established, particularly for delivery duties, in a wide variety of industries. The manufacturers, however, considered that an improved suspension system would enhance the performance of vehicles operated overseas in open country or on rough tracks and also in the more remote rural areas in Britain where a substantial proportion of delivery round mileage may lie over farm roads or off the roads. Accordingly, the decision was taken to equip the vehicle with independent front-wheel suspension. The problem was thoroughly investigated, various schemes were projected, and eventually design work was commenced on a trailing link type of suspension with a quarter-elliptic spring, so arranged as to give, in effect, a parallel-link movement and consequently, a negligible variation in castor and camber angles.

Vehicles were fitted with suspensions to this design and submitted to rigorous tests by the manufacturer's development staff and also, under regular service conditions, by commercial users. Some development modifications were made and the vehicles then underwent prolonged testing over pavé and washboard tracks at the Motor Industry Research Association proving grounds. The final design is now in production and independently sprung Trojan vehicles will be publicly exhibited for the first time at the Scottish Motor Show this month. Whilst at present only the two normal-control chassis are independently sprung, eventually all Trojan vehicles, including the standard and long semi-forward control versions, will be so equipped.

Although initially the incentive to change to independent front suspension was to improve performance over rough tracks, in particular for overseas markets, operators in Britain report other benefits. Due to the improved riding, road-holding, steering and braking characteristics, a higher average operating speed can be maintained without fatigue to the driver. It follows that the mileage of delivery rounds can be increased or existing rounds can be completed in less time. Furthermore, fragile or friable goods are less liable to suffer damage and certain foodstuffs, bakery goods, ice cream, and industrial mixes or compositions, are less likely to "settle" or become impaired since vibration is much reduced.

The improved riding qualities were well demonstrated in immediately succeeding runs in personnel wagons on a descending rough unmade road. On the first, in a standard, normally loaded vehicle with a beam front axle, the run was made in considerable discomfort at speeds that could not safely exceed 12 to 15 m.p.h. The following run under precisely similar conditions of route and loading was accomplished in relative comfort at a steady 25 m.p.h.

As regards the design of the suspension, the geometry is unexceptional. The box-section trailing link, fabricated from two $3\frac{1}{4}$ in $\times 1\frac{1}{4}$ in channels, is pivoted in bearings bolted to the forward end of the chassis side member. At the other



General arrangement of trailing link type independent front wheel suspension on Trojan 1-ton chassis. Drawing shows off-side application. Anchorage is arranged on a stout gusset member at the forward extremity of the chassis frame

Personnel wagon on the Trojan Senior chassis with independent suspension. The power unit is the Perkins P3 144, three-cylinder diesel engine developing 41 b.h.p. at 2,400 rev/min

end the pivot axle is free to oscillate in two bushes, one of which is of the screw-threaded type to provide axial location. The pivot axle has two depending arms, in the extremities of which is carried the eye-bolt for the quarter elliptic spring. This spring is built up of twelve square-cut leaves of 2 in × $\frac{3}{16}$ in section, anchored to a frame bracket at the forward end and with the two lower leaves rolled to form the eye which is fitted with a flexible bushing.

In elevation, link and spring are parallel but in plan they are arranged at divergent angles, the link being inclined at approximately 6 deg and the spring 10½ deg to the chassis side member. These angles do not affect the geometry of the system and are arranged mainly for constructional convenience and to permit adequate wheel movement on lock. The wheel is mounted on a conventional king pin assembly cottered in the end of the pivot axle at an angle of 7½ deg in a vertical transverse plane and at a castor angle of 1 deg 5 min.

Assembly of the off-side suspension unit on the chassis. Telescopic hydraulic shock absorbers are fitted to the front wheels but not to the rear axle

All wheel forces are transmitted through the link to the chassis frame and the spring is required to accept only suspension loads. Telescopic hydraulic shock absorbers are provided between the trailing link and the frame. The spring has an effective length of 18 $\frac{3}{16}$ in and the total movement of the trailing link is approximately 14 deg from load to rebound.

The chassis is produced in two lengths, with standard and long wheelbases at 7 ft 10 in and 9 ft 4 in respectively. Turning circles are 38 ft and 44 ft and the common ground clearance is 9½ in. The power unit is the Perkins P3 144, three-cylinder diesel engine, developing maximum power of 41 b.h.p. at 2,400 rev/min and delivering a maximum torque of 101 lb-ft at 1,200 rev/min. C.A.V. fuel injection equipment is used, with pneumatic governing.

Transmission is by a Borg and Beck, 9 in diameter, spring-plate clutch to a three-speed and reverse gear box, of Trojan manufacture, with synchromesh on second and third. The gear-change lever is centrally mounted. An alternative four-speed and reverse unit, without synchromesh, is available. From the gearbox the drive is transmitted by a Hardy Spicer propeller shaft equipped with needle roller universal joints to the ¾ floating rear axle. The hypoid axle drive has a ratio of 4-1.

Lockheed hydraulic brakes are fitted and the hand brake, centrally positioned in the driving cab, provides mechanical operation of the rear brakes only. Steering is by a double-roller type gear and can be arranged for either right-hand or left-hand drive as required. Five-stud fixing wheels are fitted with 6.50 × 16 in diameter, 8-ply tyres. The fuel tank, having a capacity of 8½ gallons, is mounted on the near side of the straight-sided chassis frame.

Overall dimensions of the vehicles fitted with the standard van bodies, of metal framed construction with metal and glass-fibre reinforced plastics panels, are as follows, for the standard wheelbase version. Figures for the long wheelbase van are bracketted. Length 14 ft 2 in (15 ft 9 in), width 6 ft (6 ft 4 in), and height 7 ft 10 in (8 ft 4 in).

Internally the bodies have a length behind the driver 7 ft 3 in (8 ft 6 in), width at the waist, 5 ft 10 in in each case, and height at centre, 5 ft 3¾ in (5 ft 9 in), giving loading capacity of 210 ft³ (280 ft³).

Exhaust Gas Treatment

American Investigations into Methods for Removing Hydrocarbons

In July, 1955, the Vehicle Combustion Products Subcommittee of the Automobile Manufacturers Association organized an exhaust system task group to study methods of removing hydrocarbons from automobile exhaust gases. The assignment was limited to treatment of the exhaust after discharge from the engine. The group was also asked to investigate other problems associated with exhaust treating devices, such as safety, cost adaptability, maintenance and inspection aspects.

When the group was formed, it was believed that exhaust treatment would be a long-term investigation because of the many technical problems involved, but that a greater reduction in hydrocarbon emission could be attained ultimately by exhaust treatment than by induction system devices. Events of the past two years have confirmed these two opinions. The group is made up of George J. Nebel of General Motors Research Staff; C. E. Burke of American Motors Corporation; C. M. Elliott and R. P. Graham both of Chrysler Corporation; R. T. VanDerveer of Ford Motor Company; W. J. Pelizzoni of Mack Manufacturing Corporation; and R. D. Randall of Stewart-Warner Corporation. The results of preliminary tests were recently presented to the Society of Automotive Engineers Inc.

The first activity of the group was to survey the various methods of removing hydrocarbons from exhaust gases and select the more promising ones for further study. These methods are listed in Table I. The conclusion reached, in agreement with that of the Air Pollution Foundation*, is that the most practical way to separate hydrocarbons from exhaust gases is by oxidation. This may be accomplished either catalytically or with flame-type combustion. In the first case, the apparatus or device is called a catalytic converter; in the second case, an afterburner. Both devices accomplish essentially the same result, that is, after air is added to the exhaust, most of the combustible constituents are burned to carbon dioxide and water. The major portion of the group's activity has been spent on the study and evaluation of catalytic convertors and afterburners. The separation of inorganic compounds, such as lead compounds, sulphur dioxide and oxides of nitrogen, has not been investigated.

To the designer of an exhaust treating device, the most important operating variables are the temperature, flow rate, and composition of the gas stream, all of which vary considerably. Some of the variables that may be expected from the typical automobile in different driving conditions, and their importance to the designer, are discussed in the following paragraphs.

Gas temperature

Exhaust gas temperatures range from below 150 deg to over 1,500 deg F, depending greatly upon engine operation and the location at which the temperature is measured. Lowest temperatures are attained at low-speed, idle operation and the highest temperatures at high-speed full-load operation. There are large differences, as great as 300 deg F, between the temperatures at the exhaust manifold and at the muffler. Exhaust temperature must be considered in the operation of catalytic converters because most oxidation catalysts suitable for this application do not become active until a threshold temperature is attained. When a high temperature catalyst is used, the converter should be located as near to the engine as practicable, and the connecting piping should be well insulated. These steps will conserve heat, reduce the warm-up period, and increase overall efficiency. Exhaust temperature

* *APF Report No. 10 and "Methods and Devices for Controlling the Hydrocarbon Control of Exhaust Gases", SAE Seminar on Fuels and Lubricants, Los Angeles, April 16, 1955.*

TABLE I. EXHAUST TREATMENT METHODS

Method	Comments
A. Physical separation	
Absorption	Bulky equipment. Efficiency of absorbents, non-volatile organics or aqueous solutions of oxidizing agents or emulsifiers, is questionable. Absorbent would require periodic regeneration and/or replacement.
Adsorption	Equipment simple and inexpensive. Efficiency probably satisfactory, but retentive capacity of best adsorbent, activated carbon, low. Reasonably sized adsorber will require too-frequent regeneration or replacement.
Centrifugal	Equipment complicated and expensive. Efficiency questionable.
Filtration	Not effective for gas phase separations.
Refrigeration	Expensive, bulky equipment with high power consumption. Difficult to handle large amounts of water collected. Extremely low temperatures, perhaps—100 deg F, required for efficient separation.
B. Chemical methods	
Oxidation	Efficiency theoretically very good. Equipment less bulky and complicated. Both catalytic and flame-type oxidation possible.
Addition of chemicals to exhaust	Efficiency questionable since hydrocarbons are inert. Difficult to meter proper amount of reagent.
C. Miscellaneous methods	
Air dilution	Does not accomplish a real reduction, although there may be a lessening in noticeable odour.
Recycling exhaust	Efficiency questionable. Difficult to control so engine operation is not adversely affected. May promote intake system deposits.

is also believed important in the operation of an afterburner. It seems advantageous to have the gas temperature as high as possible to increase the flammability of the exhaust-air mixture.

Exhaust flow rate

It is difficult to measure exhaust flow directly. However, it is possible to measure engine air consumption and, from this, exhaust flow can be estimated satisfactorily. For all practical purposes, air consumption at deceleration and at idle are equal. To estimate exhaust flow from air consumption, three factors should be taken into account. These are the temperature and pressure in the exhaust system, and the molar ratio of exhaust to air. Temperature is the most important of these three factors. The actual volumetric flow of exhaust gas at the conditions of temperature and pressure existing in the exhaust system is one and a half to three times the engine air consumption in standard cubic feet.

Flow rate is important to the designer because it affects the size of the exhaust treating device. If the size is kept within reasonable limits, residence time—that is the time available for oxidation—will be short, about a few tenths of a second. The wide range in flow rates also complicates the problem of adding the proper amount of required supplementary air to the exhaust.

Exhaust gas composition

The main combustible constituents in exhaust gas are carbon monoxide, hydrogen, and hydrocarbons. Concentrations of carbon monoxide and hydrogen depend primarily on air : fuel ratio, and range from essentially zero to about 13 and 7 per cent respectively. The quoted values are those at 10 : 1 air : fuel ratio. Concentration of carbon monoxide and hydrogen are (theoretically) zero at and beyond the stoichio-metric ratio, about 15 : 1. Hydrocarbon concentration depends primarily upon driving cycle. Typical values are shown in Table II. Except perhaps at deceleration, most of the heat of combustion of exhaust gas is accounted for by carbon monoxide and hydrogen.

Gas composition is important because the designer should known how much air to mix with the exhaust to burn all the combustibles. The stoichio-metric air requirement of an exhaust burning device depends only on carburettor air : fuel ratio and is expressed as the molar ratio of air to exhaust on a dry basis. Under unusually rich conditions as much as 60 per cent fresh air may be required, although the normal range is from zero to 30 per cent.

It may be desirable to add more than the minimum amount of air in order to maintain temperatures below critical limits and avoid metallurgical failures. As an example, a case where exhaust containing 13 per cent CO and 7 per cent H_2 (corresponding to an unusually rich air : fuel ratio of 10 : 1) is diluted with the minimum amount of air required for complete combustion, and the initial temperature is 300 deg F, should be considered. Theoretically, with no heat losses, the temperature after combustion would be above 1,800 deg F. In fact, the final gas temperature would be somewhat less, but still may reach or exceed 1,500 deg F, high enough to cause the designer concern in his choice of structural materials. By adding more than the minimum amount of air, the temperature can be reduced to more reasonable levels.

The presence of lead compounds in exhaust gas is also important to the designer of catalytic converters, since these compounds are notorious catalyst poisons. At present it seems more practical to develop a lead-resistant catalyst than to attempt to remove lead compounds from the exhaust. Other compounds which may be of interest include oxygen (chiefly as this affects the amount of fresh air required), sulphur dioxide, and phosphorus and alkaline earth compounds.

TABLE II. EXHAUST GAS HYDROCARBON CONCENTRATIONS FOR A TYPICAL AUTOMOBILE

Driving cycle	Hydrocarbon concentration mole per cent carbon*
Idle	0·9
Cruise	
30 m.p.h.	0·7
40 m.p.h.	0·5
Acceleration	
0–25 m.p.h. in 12 seconds	0·7
Deceleration	
30–0 m.p.h. in 10 seconds	1·6
50–20 m.p.h. coast	3·6

**Measured with Liston-Becker Model 15 analyser calibrated with N-hexane.*

Only afterburners and catalytic converters have been investigated. All devices, except the ignition cut-off burner, have been either experimental models or commercially available products. Policy has been to have one automobile company do the preliminary testing, with their results reviewed periodically by the group. If a device shows promise, it is then tested by several member companies before final group recommendations and conclusions are made.

The efficiency of a device is defined as the reduction in hydrocarbon concentration expressed as per cent. The effect of dilution air is taken into account. Hydrocarbon concentrations have most frequently been measured with the Liston-Becker Model 15 infra-red analyser. Unless otherwise noted, all tests reported in this article were made on passenger cars using commercial leaded petrol.

The devices tested so far in the industry programme are discussed below. Many of the tests were exploratory in nature, with emphasis on some single aspect, such as catalyst life, rather than overall performance. For obvious reasons, manufacturers' names or trade names are not reported.

Device A is a commercially available afterburner. Installed on a dynamo-meter test stand, the afterburner was found to be very efficient when the exhaust gas-air mixture could be ignited. This occurred only at certain operating conditions, as shown in Table III. The composition of the exhaust, and also the temperature, appear to be the most important factors determining whether or not the afterburner will ignite the exhaust-air mixture. Exhaust composition is markedly affected by carburettor air : fuel ratio.

These dynamometer tests suggest that, when installed on a vehicle, an afterburner will be operative only part of the time, probably during the idle and deceleration cycles. If so, the afterburner is less attractive as a practical exhaust treating device than the catalytic converter because (theoretically) the latter is operative during all driving cycles. It is understood that the manufacturer of this afterburner is developing a smaller sized model for passenger cars and this device will be tested when available.

Device B, the ignition cut-off burner, is a special type of afterburner designed to operate only during deceleration. In its simplest form, it consists of a spark plug placed in the engine exhaust pipe, plus control apparatus. The control apparatus is a vacuum switch and relay to connect the burner spark plug to the spark coil, and at the same time to disconnect the distributor so that the engine will not fire. The vacuum switch is actuated when the intake manifold vacuum rises above a pre-set value, about 21 in Hg. Thus, fresh fuel-air mixture passes unchanged through the engine into the exhaust pipe where it is ignited. The theory of the device is that, under high intake manifold vacuum conditions, combustion in the exhaust pipe is more complete than combustion in the engine, since the gases in the exhaust pipe are not diluted by residual exhaust.

Tests showed that the device as described above was

TABLE III. PERFORMANCE OF AFTERBURNER DEVICE A

Engine operation	Efficiency—per cent HC removed
Idle	90 to 95
1,000 r.p.m. full throttle	0
1,000 r.p.m. part throttle	0
1,000 r.p.m. closed throttle	98 to 99
2,000 r.p.m. full throttle	80

effective and that it reduced hydrocarbon concentration during deceleration to approximately the same level as that found at road load conditions. However, a popping noise associated with its operation was objectionable. Attempts were made to eliminate the intermittent explosions believed to be responsible for this noise by stabilizing the flame at a fixed point. Of the many flame arresters tried, the one most successful in suppressing noise was a close-fitting heat riser valve placed a short distance upstream from the burner. Normally open, this valve was closed during deceleration by a solenoid-actuated relay. While this flame arrester reduced noise, it also reduced efficiency to low levels. High-speed photographs showed that the flame arrester did not stabilize the flame as hoped, but simply reduced the amount of gas burned per explosion, and increased the frequency of explosions. Repeated attempts to obtain efficient, quiet burning by modifying design were not successful.

Another difficulty with this device was that no economical means was known to make it safe in the event of failure. Without some protection, it was feared that an explosion severe enough to damage the exhaust system might occur if the burner plug become fouled.

Device C is an experimental catalytic converter employing a precious metal catalyst. The test unit showed no effectiveness whatsoever. Inspection and bench testing revealed that the catalyst had been poisoned by lead compounds. Apparently, this happened after only a few miles of driving.

Device D is a commercially available catalytic converter employing a precious metal catalyst. Initial efficiencies were as high as 28 per cent, averaged for all driving cycles. After 1,000 miles the efficiency was approximately half the value. Performance was poorer than expected because of the difficulty of installing the converter on the automobile in a completely satisfactory manner. One of these units experienced mechanical failure in the aspirator section. The manufacturer is attempting to develop a catalyst more applicable to automotive use.

Device E is also a catalytic converter, but is a commercial product designed for non-automotive use. In the type of service for which this converter was designed, it has been used successfully with leaded fuel, a feature considered important. The device was installed on a test car to gain experience and knowledge, although the manufacturer pointed out that it was not designed for the type of service to which it was to be subjected in the test.

Initial performance was only fair. The convertor was effective in oxidizing carbon monoxide, but less effective for hydrocarbons. (This characteristic was built into the catalyst in view of its originally intended use.) Average efficiencies of about 40 per cent were attained initially. Tests after 1,500 miles showed some loss of efficiency.

Device F, an experimental catalytic converter, is still being tested. The manufacturer claims that the device will be at least 75 per cent efficient over the life of the catalyst forecast as 12,000 miles, and that it can be used successfully with leaded petrol. Catalyst composition has not been disclosed, but it is stated to be of non-critical materials, not platinum or palladium.

Tests were made at 1,000 and 3,600 miles respectively. Efficiency, which correlates reasonably with catalyst temperature, is slightly lower at 3,600 miles than at 1,000 miles, but is still generally greater than 80 per cent. The manufacturer stated that it is normal for the catalyst activity to decline slightly after the first thousand or twelve hundred miles, and then remain very nearly constant for the rest of its life.

Catalyst temperature, an important operating variable, depends largely upon how the vehicle is being driven, and also depends on the kind of driving to which it has been subjected in prior periods. Slow, steady speed driving results in the lowest catalyst temperatures, about 800 deg F. The other extreme is high-speed stop and start driving, when temperatures between 1,200 and 1,300 deg F are attained. The normal city driving range is from 900 to 1,000 deg F with corresponding efficiency being 85 per cent. Approximately 15 minutes of normal driving are required to bring the catalyst to its operating temperature. During this time the device is partially effective.

At the time of reporting, the test car has been driven in transportation service just over 4,000 miles with the converter installed. To date there is no indication that the catalyst has been poisoned by tetraethyl lead combustion products. The test will continue for 12,000 miles.

Engineering problems

Numerous engineering problems have been observed whilst testing these devices. Experimental models tested so far have not silenced engine exhaust noise satisfactorily. At present, it is not known whether special silencing characteristics can be built into this type of device or whether auxiliary mufflers are needed. An additional source of noise, in some instances, has been the aspirator which supplies the air needed for combustion.

It seems desirable to regulate combustion air automatically in such a way that the catalyst temperature is maintained in the best operating range regardless of how the automobile is driven. Proper control will also shorten warm-up time and minimize the possibility of the device overheating. Each device must also be well insulated, both to improve its own performance and for the comfort of passengers. The effects of an extra under-chassis heat load on brake fluid lines and on fuel lines must also be considered.

Catalytic converters are relatively large devices and some space problems have been experienced in installing them on automobiles. These problems are expected to become more critical with future automobiles and the details of converter design will have to be explored thoroughly.

Detailed cost estimates cannot be made until the above-mentioned problems are resolved. Indications are, however, that catalytic converters will markedly increase the initial cost of the exhaust system. In addition, catalyst regeneration or replacement will be required periodically. Similar problems concerning heat, noise, and installation are expected with afterburners. Detailed comments cannot be given, however, since group experience with afterburners has been limited to dynamometer testing.

Summary

Oxidation appears to be the most practical method for removing hydrocarbons from exhaust gases. A number of oxidation devices both catalytic converters and afterburner, have been tested. One of these, a catalytic converter, has shown some promise. In road tests, after warm-up, this experimental converter removed approximately 80 per cent of the hydrocarbons from exhaust gases. The catalyst has so far been resistant to poisoning by tetraethyl lead combustion products. Life tests have not yet been completed. Afterburners also show some promise, but none suitable for automobiles has been developed and submitted to the group. Indications are that the overall efficiency of afterburners will be somewhat less than that of catalytic converters.

METAL CUTTING RESEARCH

Some Activities of the Production Engineering Research Association

THESE notes give a short survey of some researches into cutting tools carried out by the Production Engineering Research Association, Melton Mowbray, together with some examples of research results. As the cutting tool is only one of many important factors in a machining process, brief reference is also made to investigations into factors such as machine tools and cutting fluids. Attention is focused mainly on drilling in order to illustrate the scope of PERA research in a particular field, but a few examples of results obtained in investigations of other types of tools are also included.

Machining research can be broadly classified into analytical and practical investigations. The purpose of analytical investigations is to determine the precise nature of the mechanism of cutting and of the behaviour of the tool and machine tool. To facilitate analysis, the machining process is usually reduced to its simplest form by turning thin discs or planing narrow strips with elementary cutting tools. High speed films, microscopic examination of the resulting chips, mathematical analysis, and other means are all used to build up a comprehensive picture of the complex phenomena that occur during machining. The practical machining techniques can then be modified if necessary in accordance with the clearer picture of metal cutting provided by the analytical investigations.

In practical investigations on the other hand, data for immediate application in the workshop are obtained without a deliberate search for an explanation of basic phenomena. Controlled tests under conditions closely allied to practice are carried out to determine changes in tool performance with variations in tool geometry, cutting speed, feed, and other factors. The results of these tests can be applied directly to specific problems of great practical importance in the workshop.

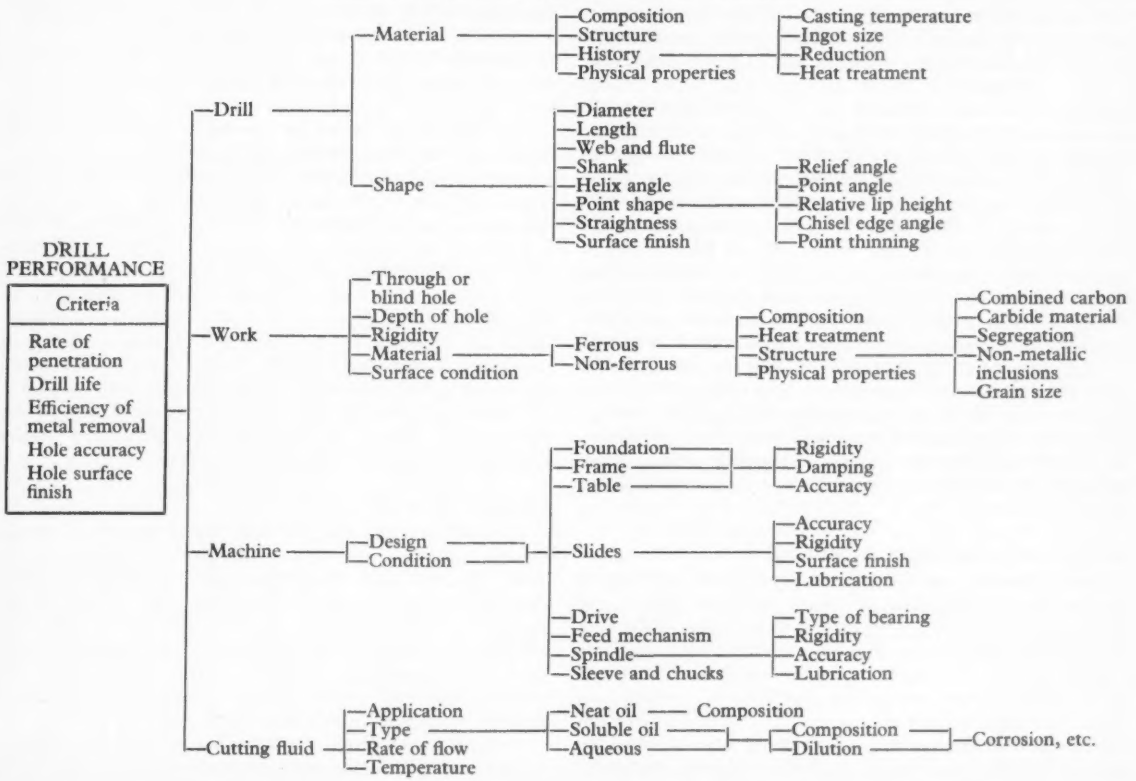
Variables and criteria in machining

An analysis of the signficant factors in machining reveals a complex system embracing the tool, workpiece, machine, and cutting conditions. An analysis for a drilling operation is shown in Table I, which also indicates the technical criteria by which drill performance is assessed. To secure the maximum improvements in the economy and speed of machining it is essential to identify these criteria and determine their significance under various conditions.

The basis of scientific method is the reproducible experiment, which, in the case of cutting tools necessitates: (a) accurate measurement of tool shape, tool life, cutting forces, surface finish, etc.; (b) close control of variables such as

""Some Experiments on the Influence of Various Factors on Drill Performance" by D. F. Galloway. Presented at Semi-Annual Meeting of the American Society of Mechanical Engineers, 19th June, 1956.*

TABLE I. ANALYSIS FOR DRILLING OPERATION



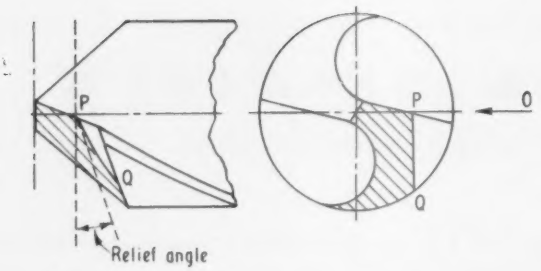


Fig. 1. Principle of optical sectioning for measuring drill relief angle

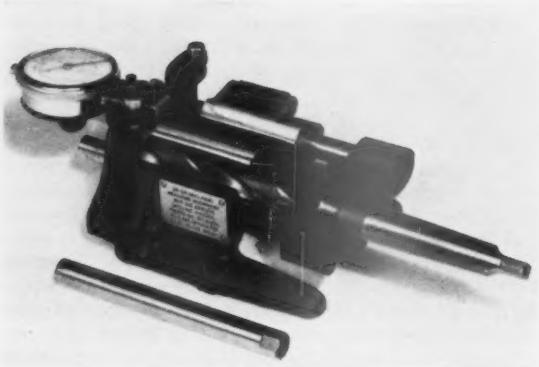


Fig. 2. Drill point inspection gauge

Fig. 3. Rigid portal frame machine used by PERA for drilling research

cutting fluids, materials machined, etc.; and (c) systematic variation of selected technical factors.

Although most cutting tools are simple in conception, their geometry is often complex, and special equipment is generally needed for the accurate grinding and inspection of tools. For example, an optical instrument based on the principle of optical sectioning has been developed for the rapid measurement of the relief angle of drills. The principle of measurement can be explained by reference to Fig. 1, which shows the method of determining the relief angle at a point P on the cutting edge. An optical section of the drill point is made in the plane PQ normal to the radius of the drill through P and parallel to the axis of the drill. The optical section is produced by projecting the outline of an accurate knife edge on to the surface of the drill, which is viewed through a microscope on the direction O in line with the radius through P. The relief angle, that is, the angle between the tangent to the curve PQ at P and the diametral plane through P is then read directly on a protractor fitted in the microscope eyepiece. Instruments developed for research are frequently re-designed for use in production workshops to facilitate effective application of research results; some examples of workshop equipment based on research instruments are shown in Fig. 2.

Control of experimental conditions

Where possible, control of the experimental conditions in cutting tool investigations is based on direct quantitative measurement. This applies not only to tool form and shape, but also to cutting fluid application, speed, feed, machine deflection, etc. Cutting tool performance may be affected by variations in the design and condition of machine tool slides, bearings, motors and frames. Undesirable variations associated with the machine tool are prevented wherever possible by the use of special equipment, such as the portal frame drilling machine illustrated in Fig. 3. The rigidity and symmetry of the frame of this machine eliminate the disturbing effects of the large deflections which sometimes occur in conventional drilling machines.

Variations in the experimental conditions can be reduced by such obvious precautions as manufacturing tools for the same experiment from the same bar of high speed steel and in the same batch. Even so, significant differences in tool life are frequently found both between different tools in the same batch and in different parts of the same tool as it is reground. Test billets can be uniformly heat treated before machining, but differences in machining properties are still found to persist between and within the billets. Variations also occur in the shape and physical properties of tools.

The effects of variations of this type can, however, be controlled to some extent by the use of experimental designs based on statistical principles. Physical and metallurgical tests are first used to indicate the probable distribution of the variations in the properties of the tools or work materials, and then, by a suitable allocation of test conditions to the test tools or billets, the effect of these variations can be substantially eliminated from the comparisons.

Duration of tests

Tool performance can be determined in tests of short, medium or long duration. The shorter the tests, the lower the cost and consumption of tools and workpiece material, but as the tests are progressively shortened by raising the machining rate, the relationships between tool life and tool shape, etc., observed in the laboratory may become less applicable to normal workshop conditions. In practice, the test conditions are accelerated as much as possible while still providing a reliable basis for the selection of workshop machining conditions. The extent of research necessary to secure data for application in workshops may be

judged from the fact that in a series of drilling investigations 1,000,000 holes were drilled, and in tapping investigations approximately 300,000 holes were tapped.

Typical research results

To illustrate the research results obtained by PERA in cutting tool investigations, attention will be mainly focused on drilling in the remainder of this article. A summary of the investigations into drill performance has been published*, and a few typical examples of the research results will be given. In parallel with the drilling research, investigations have been carried out into the performance of taps, reamers, single point tools, etc., and a few examples of the results obtained are included.

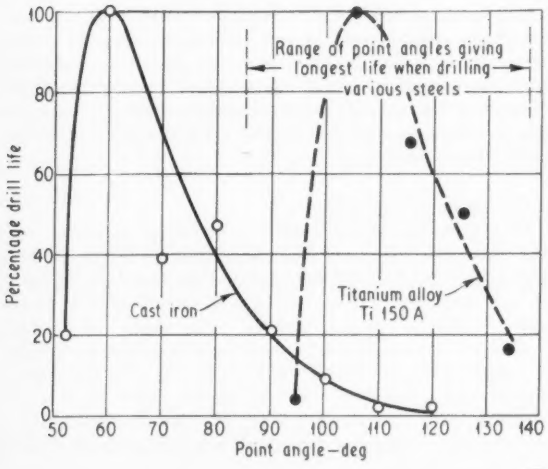
Twist drills

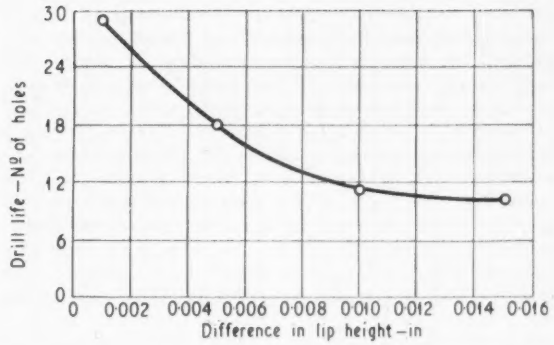
Since the modern form of twist drill machined from a solid bar came into general use nearly 100 years ago, there has been considerable progress in developing different types of drill for special purposes such as drilling deep holes, but unfortunately users generally have failed to appreciate the importance of correctly specifying and grinding the drill point, with the result that even today low standards of drill performance are accepted as a matter of course in many companies. To assist industry in reaching optimum levels of drilling efficiency, extensive research has recently been carried out into the influence of drill geometry, cutting conditions, machine tools, cutting fluids, and other factors on drill life, hole accuracy and alignment, cutting forces, and chatter vibrations. These researches have thrown new light on the fundamental reasons for some common difficulties in drilling.

Effect of drill point shape

The geometric features of a drill may have a marked effect not only on tool life, but on rate of production, power consumption, hole accuracy, and surface finish, although these effects vary substantially under different conditions. The improvements that can be achieved in drill performance by altering the shape of the drill point to suit the material drilled are indicated by the research results summarized in Fig. 4. Actual test results are shown for cast iron and titanium alloy, and the range of point angles giving optimum drill life when machining the various steels included in the investigation is also indicated. Although drills are commonly ground to a point angle of 118 deg irrespective of the material drilled, Fig. 4 shows that the point angle giving the longest life varies considerably with the material drilled.

Fig. 4. Effect of drill point angle on tool life when drilling cast iron, titanium alloy and various steels





Drill diameter ⅜ in, speed 760 r.p.m., feed 0·018 in/rev, depth of hole 1¼ in, coolant soluble oil, material axle steel

Fig. 5. Effect of difference in height on drill life

For the tests on cast iron ⅜ in diameter holes were drilled at a speed of 154 ft per min and a feed of 0·010 in per rev. Under these severe conditions drills with a point angle of 120 deg failed after a few holes, whereas 60 deg point angle drills had an average life of more than 600 holes. Although random variations in machining conditions may prevent gains of this magnitude being achieved in production operations, substantial improvements in drill performance have been reported by firms who have applied PERA recommendations for point shape. As the most suitable point shape for drilling cast iron depends partly on the drilling conditions and partly on the iron drilled, various recommendations covering a range of conditions have been made by PERA following another series of tests carried out in the Association's laboratories and in members' machine shops.

Preliminary tests have indicated that different methods of point thinning which are being investigated by PERA may significantly affect performance. Confirmation of these results is being sought in a further series of tests.

Relative lip height

Drill life and hole accuracy are markedly dependent on the symmetry of the drill point. A common error in symmetry due to grinding inaccuracies is relative lip height, as a result of which one lip or cutting edge is in advance of the other. The unbalanced forces on a drill with large relative lip height tend to displace the general axis of the drill from the axis of rotation towards a position where the resultant forces on the lips are in balance, but even so the cutting conditions at the two lips are different, and premature failure of the drill occurs. For example, Fig. 5 shows that drill life was increased nearly three times by reducing the relative lip height of ⅜ in diameter drills from 0·015 in to 0·001 in. In production drilling operations on a steel bracket, drill life was increased from 70 holes per grind to 200 holes and the penetration rate was increased from 9 in per minute to 14 in per minute by reducing the lip height of ⅜ in diameter drills from 0·010 in to 0·001 in.

It can be shown that hole oversize due to an error in lip height is very nearly equal to the relative lip height multiplied by the tangent of the semi-point angle. Calculated values are in good agreement with the experimental results; the mean hole oversize increases linearly with the relative lip height, and the oversize for a given relative lip height is less with smaller point angles.

Drilling conditions

As the cutting conditions decisively influence drill performance, the effects of feed and speed on drill life, hole

accuracy, surface finish, etc., are also investigated. Fig. 6 shows the combined effects of cutting speed and feed rate on drill life when drilling titanium alloy. If sections are taken parallel to the plane of the tool-life/cutting speed axes, curves are obtained which indicate the way in which drill life varies with cutting speed at different feed rates. Similarly curves showing the way in which drill life varies with feed at different cutting speeds are obtained by taking sections parallel to the plane of the tool-life/feed-rate axes. Of greatest interest, however, are the sections taken along lines of constant penetration rate, that is, along lines defined by the condition: cutting speed × feed rate = constant. The section ABCD in Fig. 7 has been taken in a plane representing the combination of cutting speed and feed rate giving a penetration rate of 1½ in/min. The section shows how tool life varies with different combinations of cutting speed and feed rate at a constant penetration rate of 1½ in/min. It will be observed that at this constant rate of penetration, tool life varies considerably and hence tool consumption can be reduced without affecting the rate of production merely by selecting the best combination of speed and feed.

Drilling machine vibrations

Vibration often impairs tool performance or restricts output in many machining operations, but it is particularly likely to do so in drilling operations performed on radial drilling machines, which have comparatively flexible structures possessing little damping in the lower modes of vibration. Fig. 8 shows the vibration response, in a vertical direction, of the drill head of a radial drilling machine with a periodic force of 20 lb applied vertically at the drill head. The two principal modes of vibration occurred at approximately 10 cycles per sec and 36 cycles per sec. The large amplitude of vibration at the first mode is particularly significant, as experience has shown that this is the mode which most frequently occurs during drilling.

Charts can be drawn on the basis of the results obtained

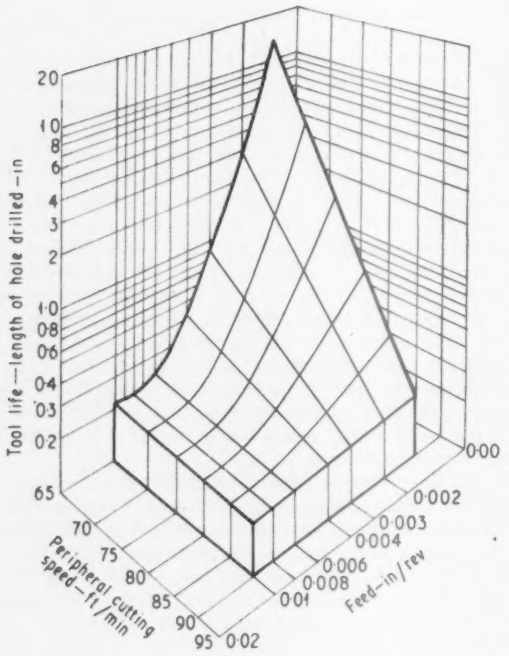


Fig. 6. Combined effects of speed and feed on drill life when drilling titanium alloy

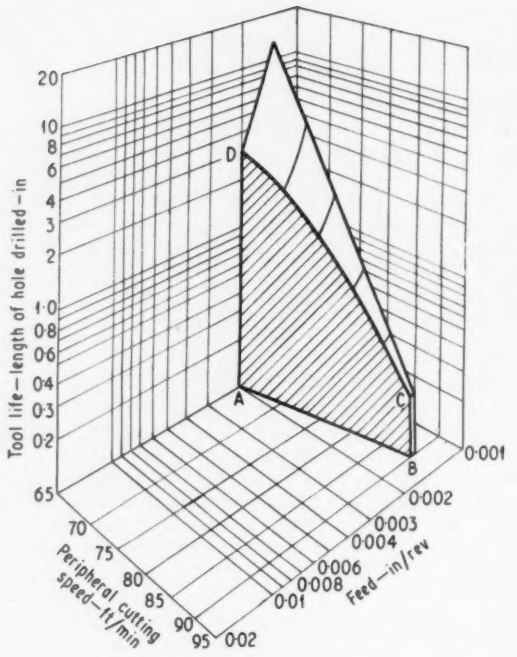
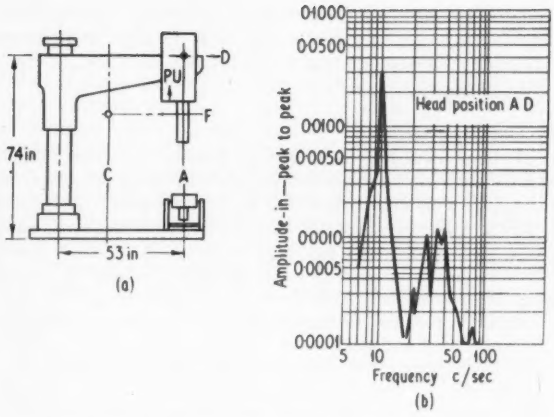


Fig. 7. Variations in drill life at constant penetration rate when drilling titanium alloy

in this type of investigation. Vibration records showing the frequency of vibration and the manner in which the amplitude builds up suggests that the vibrations may be self-inductive in origin, that is, the cutting action of the drills adds energy to the system and gives rise to progressively larger amplitudes of vibration unless the energy gain is dissipated by the damping forces of the system. To help in clarifying some of the significant features of this type of vibration, an analysis has been made of self-regenerative vibration in drilling. The main results of the analysis are shown on the stability charts in Fig. 9. The quantity δ represents the logarithmic decrement of a system comprising the drilling machine, drill, and workpiece; ρ denotes the ratio of the increase in drilling thrust to the stiffness of the system as the chip thickness increases; and n is a measure of the phase relationship between the undulating profiles of successively cut surface. The shaded areas indicate the regions of instability or chatter. Despite some discrepancies in details, the evidence of preliminary practical tests lends support to the self-regenerative theory, as indicated by Fig. 10, which shows the ranges of speeds in which chatter was observed during drilling operations on a radial drilling machine, together with the first three ranges of speed obtained from the stability charts. Tool performance can be improved by avoiding such ranges of speed.

Drilling machine deflection

The deflection characteristics of drilling machines may also have an important influence on drill performance, hole accuracy and alignment, etc. and thus observations are made of the static deflections and displacements of the heads, arms, columns, tables, frames, and bases of drilling machines. A thrust load of one ton is generally applied through the spindle of medium sized machines; this load is approximately equivalent to the thrust produced by a ¾ in diameter twist drill when cutting steel at a feed of 0·015 in per rev. The measured deflections of a typical radial drilling



(a) Position of arm and head for vibration tests
(b) Vibration response curve of drill head in vertical direction

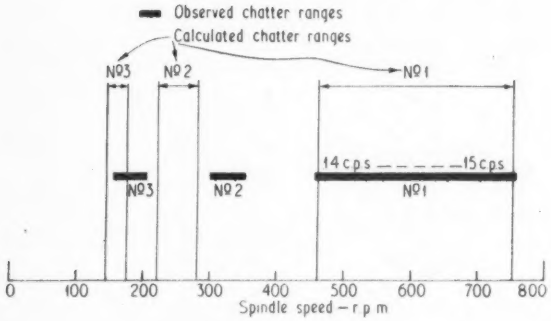
Fig. 8. Above: Vertical vibrations of drill head of radial drilling machine. Amplitude of vibration in the first mode is particularly significant, since experience has shown that this is the mode most frequently occurring in practice

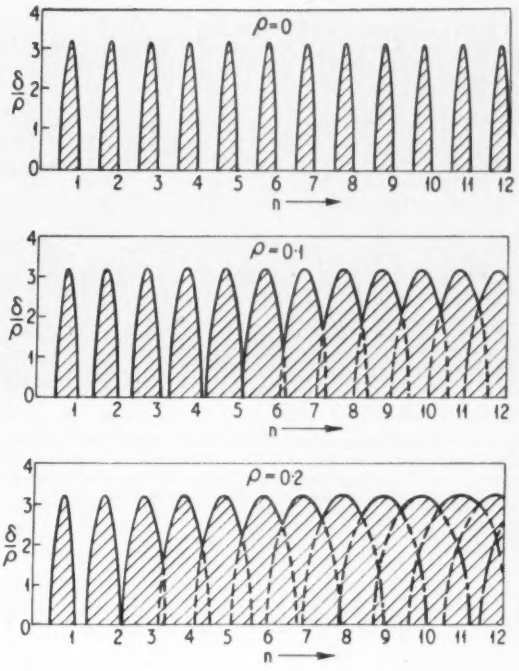
$$n=\frac{\Omega a}{4\pi n} \qquad \rho=\frac{Y}{K} \qquad \frac{\delta}{\rho}=\frac{\pi c \Omega a}{r}$$

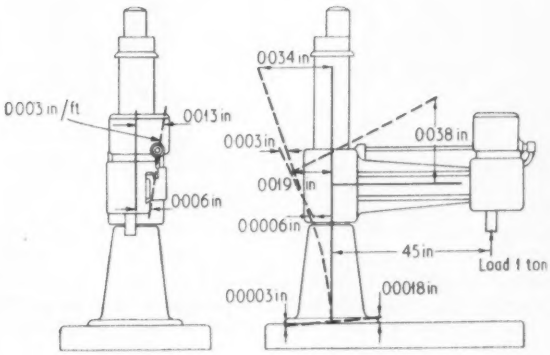
Fig. 9. Top right: Vibration stability charts

Fig. 10. Below: Observed and calculated chatter ranges for a 4 ft 6 in radial drilling machine

Fig. 11. Bottom right: Deflections of a radial drilling machine







machine are shown in Fig. 11. Each value given in the diagram is the average of three tests. The diagram is drawn with two scales along each axis, one representing horizontal and vertical deflections from initial positions, and the other the lengths of the machine parts. Thus the column is drawn in length units vertically, and deviation units horizontally, and deflections and slopes are consequently exaggerated. The principal deflections of the columns and arms of the machines tested were caused by flexure of the column.

Reamers

It is not possible here to give a full account of research into the performance of other types of tool and into the behaviour of the machine tools on which they are used, and only a small selection of results will therefore be given for other types of tool. Preliminary investigations of the influence of reamer design on the surface finish and accuracy of reamed holes indicates that hole quality can be improved by altering reamer geometry. One example of the effect of reamer shape is illustrated in Fig. 12, which shows the marked improvement in surface finish when the rake of the cutting edges was changed from —10 deg to +10 deg. Although negative rakes of this magnitude should not be encountered in practice, machine reamers often have small negative rakes which may increase surface roughness. As in the case of drills, errors in the lip height of a reamer are a common and serious cause of difficulties in reaming.

Another series of investigations has demonstrated that the surface finish and accuracy of reamed holes in low carbon steel deteriorate rapidly when speeds and feeds are increased above certain critical values. As these critical values are the limiting factors preventing the use of higher cutting speeds and feed rates in cutting operations it was considered important to determine whether improvements in performance could be achieved by changes in reamer shape. Tests have therefore been carried out to determine the effects of reamer shape on the accuracy and surface finish of holes reamed in low carbon steel. Changes in rake and circular land width influenced hole surface finish but did not markedly affect accuracy. The test results also indicated that swarf jamming has an important influence on reamer performance. Other factors investigated included chamfer angle and flute

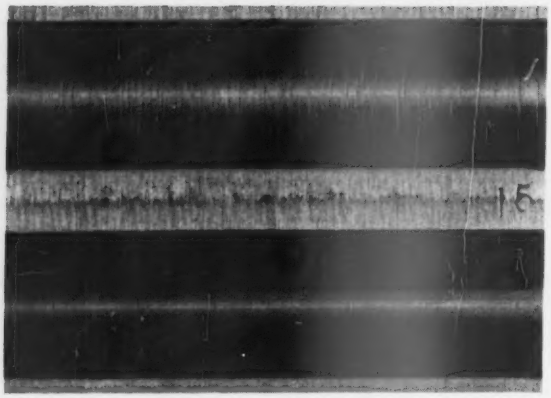


Fig. 12. Improvements in surface finish of reamed holes as a result of a change in reamer geometry. The upper hole was produced by a reamer with 10 deg negative rake, whereas the lower was produced by a reamer with 10 deg positive rake

length, and the effect of built-up edge on accuracy and surface finish, the formation of a built-up edge, and factors affecting chip-tool interface friction and temperature were also considered.

Single point tools

Typical examples of the considerable variations which may occur in the performance of lathe tools are given in Fig. 13. This comparison shows the total amount of metal removed between regrinds by a series of carbide tools used at a particular speed in rough turning operations. Tools Nos. 4 and 5, Fig. 13, could probably be used at much higher speeds than tools Nos. 1, 2 and 3, and still give acceptable tool life. An example of the combined influence of both speed and feed on the flank wear, or in other words, the life of cemented carbide tools, when machining steel, are shown in Fig. 14. Graphs of this type are valuable for indicating the combination of conditions giving the greatest economy in the use of tools.

Field investigations into problems concerned with cutting tool performance have emphasized the need for improved grinding techniques and a greater degree of control. An investigation in which conventional and unorthodox methods

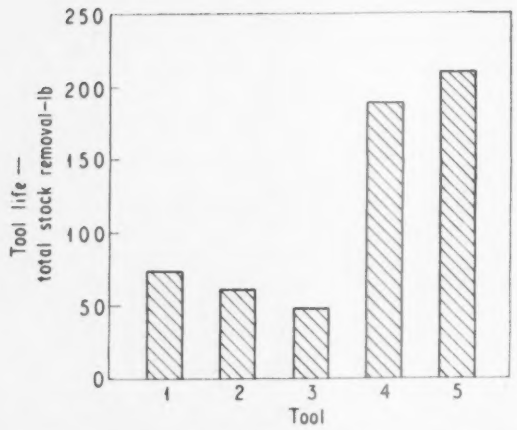


Fig. 13. Above: Variations in the performance of cemented carbide tools used in medium rough turning operations on steel

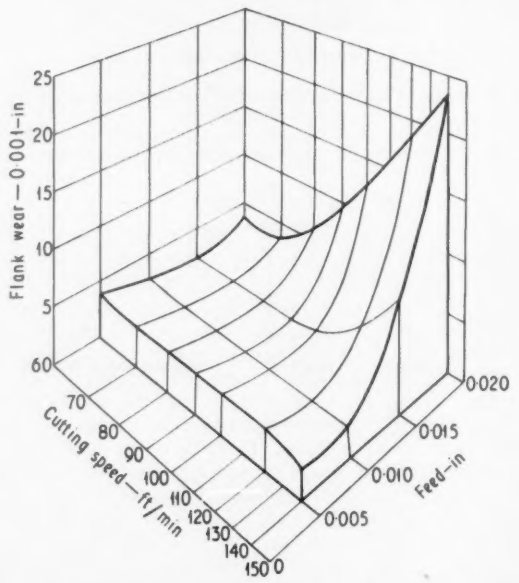


Fig. 14. Right: Average flank wear of carbide tools when machining steel at various feeds and speeds

TABLE II. SOME EXAMPLES OF ECONOMIC SAVINGS AND INCREASED PRODUCTIVITY RESULTING FROM PRACTICAL INVESTIGATIONS OF MACHINING

Changes resulting from investigation	Increase in productivity and economic savings
Improving set-up for machining operation	Saving of 11/3 per component. Tool life increased 15 times. Scrap reduced by 85 per cent
Improved method of machining drawing dies required in large numbers	Production time reduced from 53 min. to 22 min. per component. Tool life increased from 5 components to 40 components. Immediate saving of £5,000 on production
Change of cutting oils used in machine shop	Fluids cost reduced by £1,000 per annum, and tool performance improved
Modified heat treatment to facilitate machining	Machining time reduced from 51·8 to 42·3 min. Rate of production increased approx. 12 per cent, and drill life doubled. Cost reduced by about 1/9 per component
Improvement in tool grinding techniques	Tool life increased from 3 hours to 10 hours. Grinding time reduced from 20 min. to 7 min.
Substitution of finish machining operation for hand finishing and machine operations	Rate of production increased over 12 times. Cost of operation reduced by £1,300 per 1,000 components

of grinding carbide tools were studied indicated that the speed and ease of grinding associated with a particular method were of considerable economic significance. The effect of the method of grinding on carbide tool performance was also investigated.

Recommendations based on the researches are being applied in industry and savings in the form of reduced grinding wheel costs and longer tool life between regrinds have been reported. Current research includes an investigation on an automatic lathe to compare three types of high speed steel when machining alloy steel. The opportunity is also being taken during this investigation to investigate the relationship between tool wear and workpiece diameter.

Taps

In view of the many problems associated with the use of taps in industry, it is perhaps surprising that only comparatively recently has a co-ordinated programme of research into these problems been undertaken. This research has, however, already led to several important conclusions about tap performance.

In most metal cutting processes, speed, feed, and depth of cut can be varied to obtain the highest or most economical rate of production, the required finish, etc., but in tapping, only the speed, and to a certain extent, the depth of cut can be changed. The depth of cut is a particularly important variable as tapping torque rises sharply as the depth of cut increases, and hence the risk of tap breakage is greatly increased. The depth of cut is controlled by the depth of thread, and hence product designers can assist materially in reducing tap breakages in production by specifying the minimum depth of thread consistent with functional requirements. Recent research has shown that the strength of a threaded assembly can be maintained in some materials even when the depth of thread is reduced to 60 per cent.

Application of research results

As the primary objective of PERA's production engineering research is the economic gain accruing in factories from application of the research results, efforts are made wherever possible to evaluate the Association's investigations in terms of the savings made in industry. A few examples of the many economic gains in machining operations, improvements in tool life, etc., made in industry as a result of the application of PERA's recommendations are shown in Table II.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

NOVEMBER

London

Tuesday, 12th November, 6.0 p.m., at 1 Birdcage Walk, Westminster, S.W.1. Paper: "The Suspension of Internal Combustion Engines in Vehicles," by M. Horovitz, B.Sc.(Eng.), (Associate Member).

Luton

Wednesday, 6th November, 7.30 p.m. Assembly Room, Luton Town Hall. Paper: "Some Important Problems Concerning the Small Utility Car," by Dr. Ing. D. Giacosa (read by Mr. J. H. Pitchford, M.A. (Member)).

Derby

Monday, 18th November, 7.15 p.m. Midland Hotel, Derby. Address by the Chairman of the Automobile Division, R. C. Cross, (Member), entitled "Some Experiments with Internal Combustion Engines."

Scottish

Monday, 18th November, 7.30 p.m. The Institution of Engineers and Shipbuilders, 39 Elmbank Crescent, Glasgow C.2. Paper: "Servicing the Modern Vehicle," by A. D. Reid.

North-Eastern

Wednesday, 20th November, 7.30 p.m. The Chemistry Lecture Theatre, The University, Leeds. Paper: "The Ability to Stop," by H. Clements.

Western

Thursday, 21st November, 6.45 p.m. Fortt's Restaurant, Bath. A talk on "Tyre Characteristics," by T. French.

Birmingham

Tuesday, 26th November, 6.30 p.m. James Watt Memorial Institute, Great Charles Street, Birmingham. Paper: "Transmission Developments for Public Service and Heavy Goods Vehicles," by A. Gordon Wilson (Associate Member).

DECEMBER

London

Tuesday, 10th December, 6.45 p.m., at 1 Birdcage Walk, Westminster, S.W.1. Automobile Division Discussion: "Air or Water Cooling."

Coventry

Tuesday, 3rd December, 7.15 p.m. Grosvenor Room, Leofric Hotel, Coventry. Two papers from a Symposium on Superchargers and Supercharging:—"An approach to the Problem of Pressure-charging the Compression Ignition Engine," by D. W. Tryhorn, B.Sc. (Associate Member). "The Design and Development of Small Radial Flow Turbochargers," by C. A. Judson, B.Sc. (Associate Member) and E. Kellett.

North-Western

Tuesday, 3rd December, 7.15 p.m. Engineers' Club, Manchester. Paper: "Diesel Engine Lubricants: Their Selection and Utilization with Particular Reference to Oil Alkalinity," by A. Dyson, M.A., L. J. Richards, B.Sc.(Eng.), Wh.Sc., and K. R. Williams, B.Sc.

Luton

Wednesday, 4th December, 7.30 p.m. Assembly Room, Luton Town Hall. Paper: "The Suspension of Internal Combustion Engines in Vehicles," by M. Horovitz, B.Sc.(Eng.) (Associate Member).

Western

Thursday, 5th December, 6.45 p.m. The Royal Hotel, Bristol. Paper: "Pneumatic Suspensions for Motor Vehicles," by J. Sainsbury.

Derby

Monday, 9th December, 6.15 p.m. The Rolls-Royce Welfare Hall, Derby. A Joint Meeting with the East Midlands Branch. Paper: "Small Gas Turbines," by Dr. J. H. Weaving (Member).

HARPER MK.VI INVALID'S CAR

Petrol-driven and Battery-driven Models of a Vehicle Produced Exclusively For Physically Handicapped Drivers

PROBLEMS of a character quite distinct from those encountered in connection with a normal passenger car confront the designer of a vehicle intended for general use by partially incapacitated or physically handicapped persons of either sex. The variations from the mythical "standard man" are sufficiently wide and sufficiently common to cause some trouble to the builders of mass-produced cars. When to such variations are added the complications and permutations resulting from multiple injuries, incapacitating diseases or congenital malformations, the problems attain formidable proportions.

Invalid's cars are not subject to certain taxes but to enjoy this concession they must conform to certain stipulated legal requirements. They must be genuine single seaters and their legal maximum speed is 20 miles/hr. Furthermore, while vehicles are, of course, available for direct purchase by individuals, the majority are supplied by the Ministry of Health to handicapped persons for travel to and from a place of occupation, with the object of enabling them to lead a full and useful life. Being the largest customer, the Ministry of Health issues a comprehensive specification and invites tenders. Manufacturers must, perforce, design their vehicles in conformity to the strictly utilitarian requirements of the specification. There is little opportunity for styling, decoration, or equipment but in the latest Harper models an outstandingly clean and attractive appearance has been attained by the use of a moulded polyester resin-glass fibre body.

Built by the Stanley Engineering Co Ltd., The Avenue, Egham, Surrey, a member of the Harper Aircraft group of companies, these vehicles make a complete break from the tubular construction of the traditional "invalid's tricycle". The chasses frame is of channel and angle steel construction, with a light alloy beam "backbone" and a floor of mild steel sheet. Suspension of all wheels is on torsion-rubber units and the rear wheels are independently sprung. Brakes are fitted to the rear wheels only and are hydraulically operated. Complete weather protection of the occupant is provided by either hard-top or convertible bodies. The general appearance is that of a modern miniature three-wheeler and the absence of external features usually associated with invalid's vehicles has a beneficial psychological influence on the user. This applies with particular effect in the case of users who have only recently suffered a disability and is an important factor in rehabilitation.

It is an unfortunate feature of a necessarily single-seater vehicle that the handicapped learner suffers a further handicap in that he cannot be accompanied during training on the road. The usual procedure is for the instructor to explain the operation of controls while stationary and to walk alongside the vehicle during preliminary runs at the lowest speed. Subsequently the learner is launched alone on to the roads. The Stanley Mk. VI body has adequate width to accommodate side-by-side seats and the manufacturers have produced a dual-control vehicle in which the steering tiller and all controls are duplicated. The instructor at all times can exercise complete control of the vehicle and can give the learner verbal and visual instructions whilst travelling at normal speeds. Such dual control-vehicles, made specially to the order of authorities and organizations concerned with rehabilitation, are understood to be unique.

Structurally both vehicles, petrol and electric models, are the same. They differ in the power unit, transmission, and controls. They are distinguished in appearance, however, the petrol car being finished in blue and the electric car in grey. The hard top in both instances is finished off-white and in the case of convertibles the hood is of a brown Vynide plastics material. Primarily, the electrically propelled version is intended for users having only one hand capable of performing control movements. The unladen weights are approximately 5 cwt for the petrol car and 8 cwt for the electric model. The difference in weights is accounted for by the three traction batteries of the electric model.

The Harper Mk. VI invalid's car has a polyester resin-glass fibre body in either hard top or convertible versions. Petrol car is finished blue and the electric car grey

Chassis

The main frame is a flat rectangular structure forming the central section. It comprises a $1\frac{7}{8}$ in×2 in I-section beam of Hiduminium 44 light alloy arranged as a backbone and flanked by steel angles which are extended to the rear beyond the main frame to form cantilevers for the support of the power unit and transmission. Front and rear ends of the frame are closed by angled steel pressings and steel angles form the lateral margins. From the corners of the structure diagonally arranged angles are secured by through bolts to centrally positioned anchor plates above and below the light alloy beam to ensure lateral rigidity.

The body bulkheads are attached to upstanding flanges on the front and rear transverse members. At the rear the flange is additionally supported by four pressed steel brackets and from two of these, tie rods are fitted to brace the cantilevered members which are tied near their rear extremities by a transverse angle member. A sheet steel skin attached directly to the frame forms the floor panel of the vehicle. Frame assembly is, in the main, by bolting and all component items are interchangeable.

Specially built two-seater petrol cars can be supplied equipped with dual controls for driving instruction

Suspension

Suspension of all three wheels is by means of Bramber No. 6 Flexitor torsion-rubber units, approximately $8\frac{3}{4}$ in long. The spindles of these units are serrated, 60 teeth of 60 deg inclusive with root diameter of 1·144 in, for the attachment of the wheel suspension arm, so adjustment of chassis height is rendered particularly easy. Normally this height is 9 in above ground level.

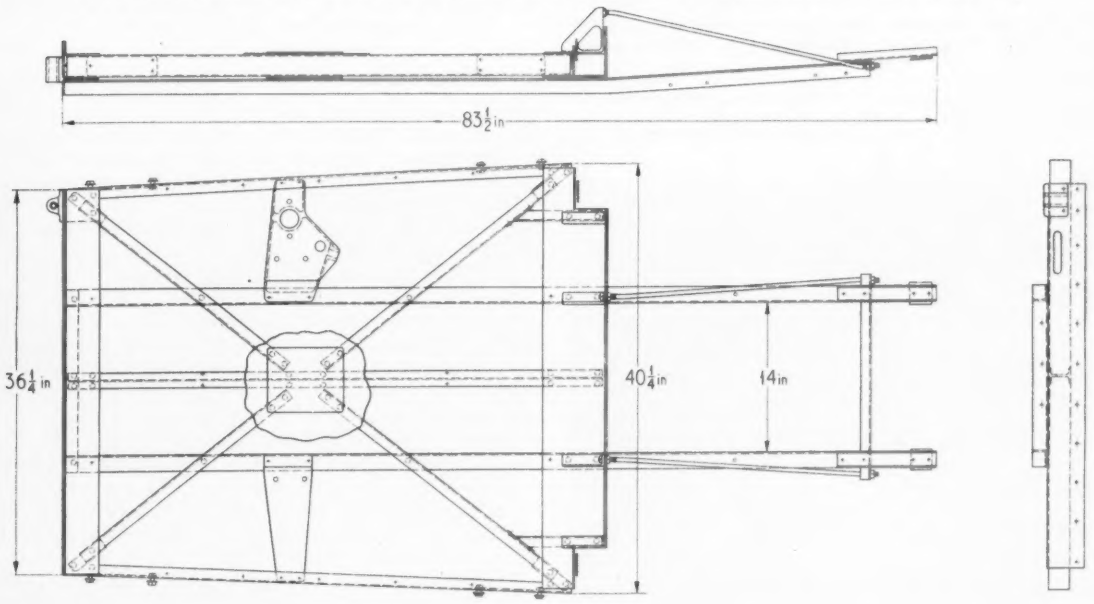
The leading type front wheel arm, bowed to accommodate steering movement of the wheel, is a forging in En 18R steel and a standard Ford king-pin and bushes are fitted at the wheel end. Castor angle of the king-pin will, of course, vary as the angle of the arm changes under load and rebound. As arranged on the chassis the angle will not, under normal conditions, exceed 3 deg. Wheel movement in suspension is controlled by an Armstrong double-acting telescopic shock absorber.

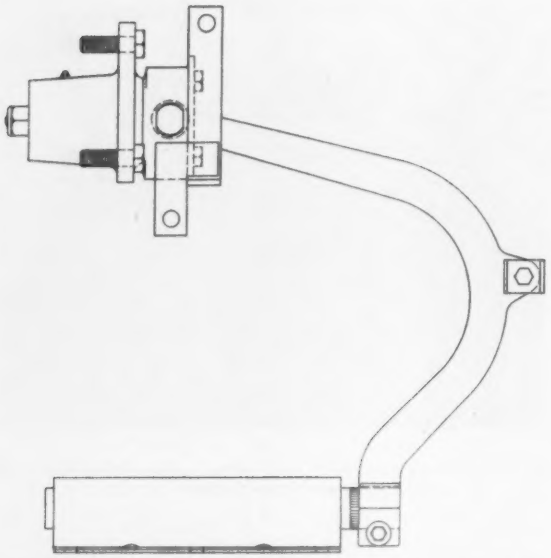
At the rear, the independently sprung wheels are carried on cranked trailing arms of spheroidal graphite iron. The wheel stub axle runs in spaced ball bearings housed directly in the free end of the arm. Drive is transmitted through a Layrub coupling splined to the end of the stub axle and secured by a Simmonds self-locking nut. Disc wheels with a three-stud fixing and shod with 20 in×4 in tyres are fitted all round.

Power unit and transmission

A blower-cooled engine and gear unit of the Villiers 9E range is installed in the petrol-driven model. Bore and stroke are 59 mm and 72 mm respectively giving a swept capacity of 197 cm³. Compression ratio is 7·25:1. The enclosed primary drive from the engine crankshaft to the clutch on the gear unit is by a $\frac{3}{8}$ in pitch roller chain giving a ratio of 2·15:1. Forward ratios in the four speed and reverse gearbox are 1:1, 1·27:1, 1·78:1 and 2·94:1. The unit is attached to the chassis by four Metalastik dual-sandwich mountings operating in shear.

Bolted assembly of chassis frame, common to petrol and electric cars. The central beam is in Hiduminium 44 light alloy



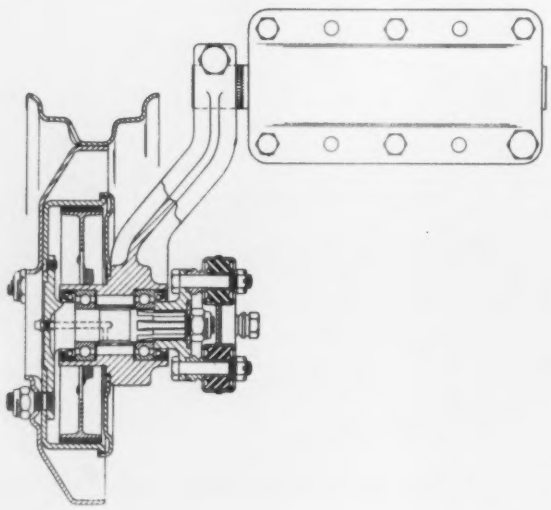


Leading-arm type front suspension. The Bramber Flexitor torsion rubber unit is bolted to the chassis frame and an Armstrong shock absorber is fitted

Gears are normally selected by a lever on the right of the centrally positioned driving seat and operating through a gate on the gear unit.

The legal maximum speed for the petrol car is 20 m.p.h. but in preference to governing the engine to keep down the speed a policy of employing relatively low gear ratios is followed. By this means acceleration and hill-climbing capacity are not impaired but are, in fact, appreciably enhanced. From the gear unit the drive is taken by a ½ in pitch roller chain giving a reduction of 1·5:1 to a simple spur gear reduction box, having a ratio of 3·75:1, mounted on the chassis to the rear of the power unit. Thence a short shaft with Layrub couplings at each end transmits the drive solely to the near-side rear wheel. This single-wheel drive is traditional on such vehicles and has been proved to be entirely satisfactory. In top gear the ratio from engine shaft to driving wheel is 12·09:1.

Trailing-arm type independent rear wheel suspension. View shows near-side wheel through which alone the drive is transmitted on the petrol car



Steering

Steering by a transversely arranged horizontal tiller on a vertical post, positioned to the right of the seat and supported on a light alloy member bridging the two off-side frame members, is by means of a link to a horizontal lever pivotally mounted in a bearing bolted to the frame front cross member. From the front end of this lever a short ball-jointed link is connected to the wheel swivel head.

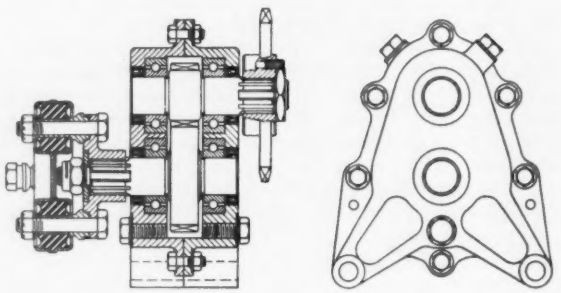
Users prefer a lighter or heavier steering effort according to their physical capacity or temperament and means to meet this demand is simply arranged by providing three alternative holes to receive the ball stud in the end of the lever. Selection of these holes will, of course, alter the leverage ratio and give what is, in effect, a higher-geared or lower-geared steering. Another consequence is a slight variation of the turning circle of the vehicle between approximately 16 ft and 18 ft; the high-gear steering gives the smaller circle.

Brakes

Braking is applied to the rear wheels only. Standard Girling hydraulically actuated brakes of 7 in diameter are fitted and are operated by depressing the tiller. For parking, the brakes can be applied by a ratchet type hand lever positioned forward of the gear lever. Ratchet release is by means of a thumb button on the end of the lever.

Body

The polyester resin-glass fibre laminate body is made in three main sections; the front end complete with the windscreen frame, the rear end housing the engine and gear unit,



On the petrol car the drive from the four-speed gear unit is by chain to a reduction gear and thence by shaft to the rear wheel

and the hard top. These are moulded on the wet lay-up method, using chopped mat and woven cloth glass fibre reinforcement. The resin mix is stippled and brushed into the laminations. One wide door only, on the near side, is called for in the Ministry of Health specification. This and the simulated door panel on the opposite side of the body are of pressed steel. It may be noted that where necessary to meet the requirements of the user, the controls can be arranged on the left-hand and the door fitted on the off side.

Access through the wide door aperture on to a flat, unobstructed floor is relatively convenient. If desired the seat, which is adjustable in fore and aft direction, can be arranged to slide transversely from its central position to the door to facilitate the transfer of the user from a wheel-chair. A folding wheel-chair can be accommodated inside the body alongside the driving seat on the nearside. Detachable side screens, to fully enclose the compartment, are stacked against the metal panel on the off side and secured against rattling by a braided elastic strap. A loose bridle strap is fitted to the door to enable the user to swing it from a wide-open to the closed position from the driving seat.

The complete body shells are produced at the Exeter works

of the C. E. Harper Aircraft Co. Ltd., and transported to Egham by road. A shell weighs approximately 70 lb and, assembled on the vehicle, is 9 ft 8 in long, 4 ft 1 in wide, and 4 ft 8½ in high overall.

Equipment and controls

The power unit incorporates a flywheel magneto for ignition and a Siba Dynastart unit for starting and lighting in conjunction with a Chloride 12V, 23 ampere-hour battery. Standard electrical equipment includes two pre-focus dipping headlamps with parking lights, two rear lamps with stop lights, winking direction indicators at both front and rear, windscreen wiper, and horn. The battery is cradled on the floor of the body underneath the facia board on the right-hand side. All the electrical controls—headlamp dipper switch, horn button, direction indicator switch, lights switch, and a combined locking ignition switch and starter switch—are arranged on a horizontal panel at approximately seat height on the right, outside the parking brake lever and the gear-change lever.

On the tiller are mounted the twist-grip throttle control, the clutch control lever, and a fuel cock lever. All three are of conventional motor cycle types operating through flexible cable connections.

Other equipment supplied with the vehicle includes two detachable side screens, two rear reflectors, speedometer, driving mirror, and, of course, a kit of tools and an instruction handbook. The fuel tank has a capacity of 3 gallons and includes a reserve supply device holding sufficient to give approximately 10 miles running.

Electric car

As mentioned earlier, the electrically driven version is constructionally the same as the petrol model, and has the same suspension system. Power is supplied by three Tungstone 12V traction batteries, coupled to operate as a 36V unit of 100 ampere-hour capacity. This is adequate for approximately 40 miles running of the vehicle at speeds up to 12 m.p.h. Each battery is 24 in long, 7¼ in wide, and 10 in high, and weighs about 1 cwt.

The 1 h.p. traction motor, a Lancashire Dynamo and Crypto unit, runs at 3,100 rev/min and drives through a worm-type reduction gear having a ratio of 17·5:1. Embodied in the worm wheel is a spur-pinion type differential gear which transmits the drive through axle shafts each equipped with two Layrub couplings to the independently sprung wheels. The gear unit is rigidly mounted on the frame cantilevers and the motor is flanged directly to the gear casing. Stays from the chassis frame steady the overhanging forward end of the motor.

Access to the engine and gear unit, or to the traction motor and batteries of the electric model, is by way of the boot lid

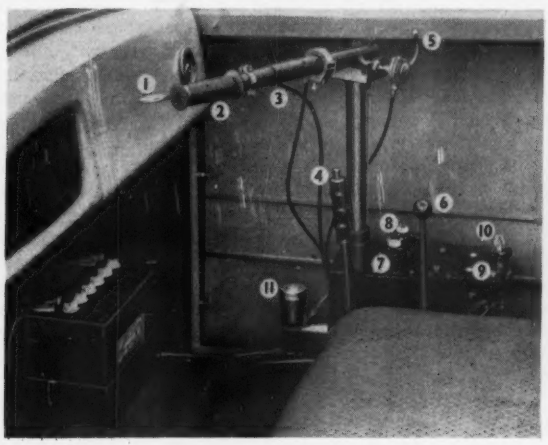
Electrical braking

In addition to the normal wheel brakes operated by depression of the steering tiller, the traction motor is utilized to provide electric braking through the transmission. This system is fully effective, in fact, two-stage switching has been found to be desirable. The two-position control gives first a gentle braking effect suitable for normal slowing and coming to rest and secondly a more powerful application for use when descending steep gradients or for emergency stops.

This system of electric braking, which is virtually essential for users suffering certain disabilities, constitutes the main reason for the change in the transmission layout from that of the petrol car. Being applied by a switching operation, it is insensitive in control and should full braking effect be switched on inadvertently under unsuitable conditions and applied to a single driving wheel, the vehicle would probably be slewed off course. The provision of two-wheel drive through a differential gear obviates that hazard.

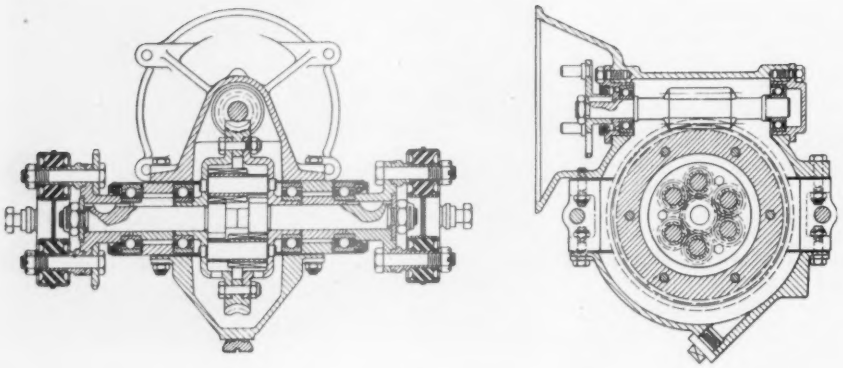
Arrangement of controls on petrol car

1 *clutch lever*; **2** *twist-grip throttle*; **3** *steering-tiller and depression brake*; **4** *parking brake*; **5** *fuel cock control*; **6** *gear-change lever*; **7** *headlamp dipper switch and horn-button*; **8** *winking direction-indicator switch*; **9** *lighting switch*; **10** *combined ignition switch and starter*; **11** *hydraulic brake cylinder*



Control layout on electric car

1 *tiller, depression brake and twist-grip control*; **2** *speed-control box*; **3** *visual indicator*; **4** *headlamp dipper switch and horn-button*; **5** *direction-indicator switch*; **6** *parking brake*; **7** *battery charger*; **8** *lighting switch*; **9** *immobilization key*; **10** *voltmeter*; **11** *charging socket*; **12** *hydraulic brake cylinder*; **13** *resistance box*



Final worm drive and differential gear assembly for the independently sprung rear wheels of the electric car

Electric controls

A specially designed twist grip on the tiller gives progressively three forward speeds when turned towards the driver and two electrical braking positions if turned in the opposite direction. Engagement of reverse drive is not possible until the grip is returned to neutral, between first speed and first braking positions. The grip can then be moved axially to the right and subsequently turned away from the driver to give first and second reverse speeds. On the speed control box mounted on the tiller is a window through which an indicator of the forward, braking, and reverse switching position currently in use can be observed. Also arranged on this control box is the headlamp dipper switch and horn button and the direction indicator switch.

An electrical control box on the floor at the right of the seat carries the master key switch by which the vehicle can be immobilized. The 12V lighting system, which is operated from the traction batteries through a transformer, is independently controlled by another switch. Also mounted on this box are the battery charging socket and the voltmeter by which the state of the batteries can be seen on depressing a button.

Equipment

The batteries are mounted on the cantilever members, one on each side of the traction motor and one transversely forward of the motor. Access to the longitudinally arranged pair is gained by raising the boot lid but the other is reached from the interior of the vehicle by way of a detachable panel in the rear bulkhead. A battery charger is housed in the electrical control box and the resistance box is carried on the floor of the compartment, well forward on the right-hand side.

Other equipment is the same as that for the petrol car, with the sole exception that a simple mileage recorder is fitted instead of a speedometer, since speeds are low.

PROGRESS IN INDUSTRY

Presentation of Social Research into Present-day Problems

THE pace of technical change today is much faster than it has ever been in the past. Its effects on industry are great and the effects on human beings may be profound. Obviously, technical change raises a host of problems, some of which it may be difficult to solve satisfactorily

Research into these problems is, of course, being continually carried out. The research workers produce exhaustive and detailed reports that can be of great value in stimulating new ideas and new methods of tackling the many difficulties. Unfortunately, the reports are necessarily too long for executive or trades union officials to read. In other words, the people they most concern and might help are too busy to spend the time to read a lengthy academic report.

This has been recognized by the Department of Scientific and Industrial Research, which has, therefore, initiated a new series of publications called "Problems of Industry." The booklets will be as short and as easy as possible to read. They will be "industrial versions" of academic reports or reviews of comparable research done independently by several teams. The idea behind these publications is to keep people in industry thoroughly informed as to what is going on, and to provide a forum for responsible new thinking and to stimulate independent discussion and action, including further research. Naturally, the conclusions and speculations in these booklets will be those of the research workers, not those of the Department.

The first of the series is called "*Men, Steel and Technical Change.*" It deals with the changes that have taken place in a steel making firm, particularly those that have taken place in the last 20 years. The problems of management, chiefly those arising from increased specialization, the relations between trades unions and management, and the attitude of employees to change, are all described. In particular, the melting shop is the subject of a detailed study.

This first booklet is a summary of a much longer report published by the Liverpool University Press. It is hoped that the booklet will make some contribution towards an understanding of what makes workers view technical change either with favour or disfavour, and why, in certain circumstances, they accept and in others resist it. The Department also hopes that the new series will help industrial firms to investigate problems—social, economic and technical—that at present are proving difficult to solve; and to see how, and how closely, these aspects are linked.

These booklets may be obtained from H.M. Stationery Office, price 2/-, by post 2/4.

The Seating of Wet Clutches

An Analysis of the Factors that Determine the Time Required for a Clutch of the Wet Disc Type to Engage Fully

Dr. P. A. Vasavada, A.M.I.Mech.E., A.F.R.Ae.S.

An important parameter, governing the efficiency of a clutch of the wet disc type is the time it takes to seat. The following method for determining this time can be applied to two lubricated surfaces having a relative velocity of approach, but no relative velocity of sliding, there being a film of lubricant between the two surfaces. This analysis can be applied to discs with a central hole, which are surrounded by oil. In a given time, the change of thickness of the film of oil, termed by Underwood[1] the *squeeze film*, between a pair of axially-loaded clutch plates is given by Archibald[2] as:

$$\Delta t = \frac{3\pi\mu}{4W}\left(\frac{1}{h_2^2}-\frac{1}{h_1^2}\right)\left[r_2^4 - r_1^4 - \frac{(r_2^2-r_1^2)^2}{\log\frac{r_2}{r_1}}\right] \qquad (1)$$

where:

Δt = time required to squeeze a film from a thickness h_1 to a thickness of h_2, sec

μ = viscosity of the oil in the clutch, lb-sec/in²

h_1 and h_2 = initial and final values of the film thickness respectively, in

r_2 and r_1 = the external and internal radii respectively of the disc, in

W = load on the disc, lb.

Equation (1) is based on the assumption that the flow of the viscous medium, as it escapes from between two surfaces, Fig. 1, is directed radially inwards as well as outwards. This expression is, therefore, applicable to determine the time to seat a wet disc type clutch. The time of seating is, of course, the time required to squeeze the film of the viscous medium or oil from its thickness in the initial unseated position to that in the final seated position.

Inspection of equation (1) shows that the time required for seating is directly proportional to the viscosity and inversely proportional to the load. It also depends on the film dimension and the radial dimensions of the disc. The extent to which each of these factors affects the seating time depends on the expressions within the brackets. Therefore, a more rational indication of the time required can be obtained by rearranging the expression to form a non-dimensional equation, as follows:

$$\frac{\Delta t}{\frac{3\pi\mu r_1^2}{4W}} = \frac{r_1^2}{h_1^2}\left[\left(\frac{h_1}{h_2}\right)^2 - 1\right]\left\{\left(\frac{r_2}{r_1}\right)^4 - 1 - \frac{\left[\frac{r_2^2}{r_1^2}-1\right]^2}{\log\frac{r_2}{r_1}}\right\} \qquad (2)$$

In this expression, the non-dimensional time on the left-hand side of the equation can now be expressed for various values of h_1/h_2 and r_2/r_1. However, before proceeding to do this, it should be noted that in a clutch disc the values of r_1 and h_1 will be given: therefore r_1^2/h_1^2 will be a constant and will be multiplied by the right-hand side of the equation. For example, r_1 may be 5 in and h_1 may be 0·0005 in, in which case $r_1/h_1 = 10^4$, which can be taken as a constant for that particular disc clutch. Thus, it can be seen that the following form will be found more convenient:

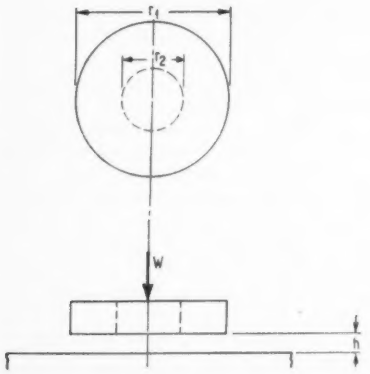


Fig. 1. Left: Diagram illustrating the geometrical arrangement on which the calculations are based. The flow of the viscous medium, as it escapes from between the two surfaces, is assumed to be directed radially inwards as well as outwards

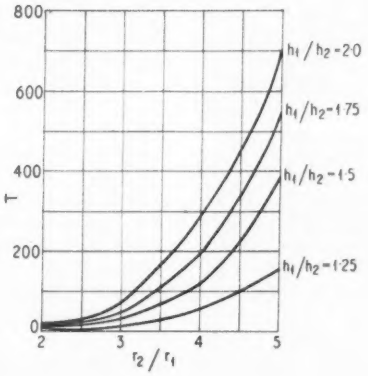


Fig. 2. Right: The non-dimensional time parameter T plotted against r_2/r_1 for four different values of thickness ratio h_1/h_2

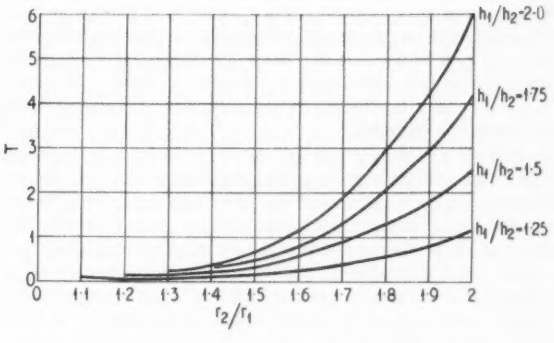


Fig. 3. Values of r_2/r_1 plotted on a larger scale than in Fig. 2.

$$T=\frac{\Delta t}{\frac{3\pi\mu}{4W}\frac{r_1^4}{h_1^2}}=\left[\left(\frac{h_1}{h_2}\right)^2-1\right]\left\{\left(\frac{r_2}{r_1}\right)^4-1-\frac{\left[\frac{r_2^2}{r_1^2}-1\right]^2}{log\frac{r_2}{r_1}}\right\} \quad (3)$$

The term T in this equation is the non-dimensional time parameter. In Fig. 2, T is plotted against r_2/r_1 for four chosen values of h_1/h_2, while Figs. 3 and 4 show the values of r_2/r_1 to a larger scale. Values of r_2/r_1 have been taken up to 1·1 because when r_2/r_1 equals 1·0 the value of time will tend towards negative infinity, the curve crossing the x axis before the origin.

These curves show that for a fixed value of h_1/h_2 the time to seat, expressed non-dimensionally, increases with the ratio r_2/r_1; it does this slowly at first, but more rapidly as the value of r_2/r_1 increases. The lower the value of h_1/h_2 for a given ratio of r_2/r_1, the smaller is the time to seat. Therefore, to obtain small seating times, the value of either r_2/r_1 or h_1/h_2 must be low. This, of course, is based on the assumption that the viscosity remains constant. However, if the viscosity is affected by temperature, its variation must be taken into account.

In Fig. 2 is shown a power curve obtained from equation (3). From this, it can be seen that the time to seat is proportional to a high power of the dimensions, namely fourth power of radial dimension and second power of the film dimension. It follows that to reduce this time to a minimum, small radial dimensions are preferable. Small values of the ratio of the external to the internal radius or, in other words, a large central hole in the disc will enable smaller seating times to be obtained. Additionally, the oil chosen for the disc clutch should be such as to give a favourable film ratio. Equation (3) can be used for rapid calculations to determine the seating time characteristics of a wet disc clutch.

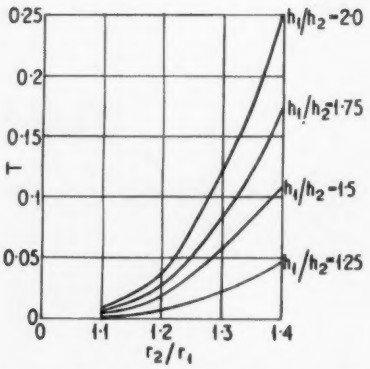


Fig. 4. The lower ends of the curves in Fig. 3 plotted on a large scale

Grateful acknowledgment is made for Dr. F. R. Archibald's helpful suggestions with regard to this work and also for the encouragement given by the authorities of the Air Force Technical College, Bangalore.

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Experimental Double-Deck Coach

AN experimental Green Line coach is now being tried out by London Transport. It is known as CRL.4—"Coach Routemaster Leyland"—and is specially designed for smooth riding. CRL.4 is the third vehicle in the Routemaster series to enter passenger service. It is similar in design to the earlier prototypes, but has mechanical units by Leyland Motors Ltd. Unlike the previous prototypes, the suspension is by coil springs only on the rear, the front being fitted with independent torsion bar suspension. The transmission is fully automatic, with an electro-hydraulically operated epicyclic gearbox controlled through a circuit recently developed by Self Changing Gears Ltd. In due course power assisted steering will be fitted. The bodywork is by Eastern Coach Works Ltd., to London Transport design and specification.

The automatic gear control, designed and built by Self Changing Gears Ltd., Coventry, is fitted to give control over the second, third and fourth gears; the first and reverse gears are manually selected. If the gear lever is placed in either second or third position, the selected gear remains engaged until the lever is moved, but on placing the selector lever in fourth (top) gear, the full automatic characteristics are brought into play, and changing up and down between second, third and fourth takes place according to the speed of the vehicle and the load on the engine.

The over-riding arrangements whereby the automatic characteristics may be put out of action and second or third gear selected at will, is a feature introduced to meet circumstances such as descending hills or moving slowly in dense traffic, in which the fully automatic characteristics would be unsuitable. Bottom gear is rarely needed for ordinary forward running of a bus in London, but is occasionally required for starting on or climbing a heavy gradient. It has, therefore, been isolated from the automatic system.

The suspension system has been developed to provide a minimum of roll combined with soft springing. Full independent suspension, as on modern passenger cars, has been provided. This is effected by means of long torsion bars anchored to the bodywork. Damping is controlled by tubular shock absorbers mounted between the wishbones, with settings specially developed for the vehicle. The rear suspension is by coil springs.

London Transport designed and developed the heating and ventilation system. Fresh air for both saloons is admitted through an aperture immediately below the front destination box; it passes to the upper saloon through five diffusers in the upper front bulkhead, and to the lower saloon through two diffusers in the centre of the lower front bulkhead. Although none of the front windows can be opened, six quarter-drop windows on the upper deck and four on the lower deck provide additional ventilation in warm weather.

The volume of air passing through the system into the lower saloon may be regulated by a control attached to the upper of the two diffusers. Apart from this control, the volume of air passing into the vehicle is dependent upon the external air velocity and the vehicle. At a speed of 25 m.p.h. 11 complete air changes per hour can be attained.

To provide draught-free ventilation it is necessary to keep air velocity in the saloons to a minimum. To this end, the size and location of the diffusers have been carefully considered. In cold weather the air is heated as it passes through a radiator under the first destination box. The heater radiator is inter-connected with the normal radiator and engine cooling system, and thus utilizes some of the waste heat generated by the engine. A heat control is located in the centre of the lower saloon front bulkhead. Control is effected by varying the flow of water through the heater radiator—no flow for cool air, increasing to full flow for warm air. A saloon temperature of 25 deg F above ambient can be obtained.

Gear Lubricants

American Developments for Hypoid Axle Arrangements

OF all the functional components of an automotive vehicle, the hypoid axle is easily the hardest working and possibly one of the most neglected units. Over the past half century and longer, engine designers have repeatedly had it impressed upon them that it is far easier to design high heat input into an engine than to provide adequate heat dissipation to prevent destructive temperatures. Similarly, it is far easier to deliver increased engine horsepower and torque to a rear axle than it is to build satisfactory, trouble-free power transmission into a critical unit which is circumscribed by size and weight limitations.

The trend towards higher horsepowers makes possible higher comfortable cruising speeds but, from the standpoint of axle life, it means increased torque and axle loads, particularly during acceleration. Lewis and O'Brien[1] have pointed out that there has been a considerable increase in direct drive through larger engines. The doubling of both engine horsepower and transmission low-gear ratio has resulted in a low-gear loading which may be several times the wheel-slip torque. Hypoid axle design has been in the direction of more hypoid pinion offset, one 1957 car of U.S. manufacture[2] having a drive pinion which is now offset 2·25 in below the centre line of the ring gear compared to an offset of 1·5 in last year. Higher offset automatically introduces more sliding between the pinion and ring gear.

Commercial vehicles have followed the same path of increased horsepower and axle loading. The axle problem is made even more difficult in commercial vehicles than in passenger cars because it is very easy to overload a commercial vehicle. In addition, as Heinen[3] has pointed out, the pinion speeds of the truck axle at 55-60 m.p.h. are equivalent to those of a typical passenger car at about 80 m.p.h. Thus, the failures through scoring are similar for both passenger-car and truck axles at these high speeds. Tests in military vehicles at Yuma, as recently reported by Streets[4] and other passenger-car and commercial-vehicle experience bear out this similarity in high-speed failure. One of the results, therefore, of the tremendous competition for more horsepower is a greatly increased burden on the rear axle.

These intensified stresses on an already critical working part place added responsibilities upon the manufacturer, the lubricant supplier, and the vehicle user. The opinion has been expressed that lubricants and lubrication have become a limiting factor in load-carrying capacity of axles. However, anti-scoring protection in passenger cars is not enough. Some car builders have found that while the active sulphur-type lubricants used as initial fill are very effective in preventing destructive metal-to-metal contact, they have been deficient in protection against wear. This leads to looseness in anti-friction bearings, excessive backlash, noisy axles, and other troubles. The excessive wear of these active sulphur-type lubricants compared to multi-purpose types has been observed on various laboratory machines. Also, while excellent in anti-weld protection, the active sulphur lubricants do not have satisfactory high-torque characteristics and, if inadvertently used in heavy-duty truck axles, may lead to scoring and ridging and eventual destruction of the gears.

On the other hand, the inactive-type lubricant used for initial fill of commercial-vehicle axles has proved unsatisfactory in some high-speed passenger-car tests, and resulted in gear scoring and early axle failure. Furthermore, since drive-pinion speeds of commercial vehicles are now in the range of high-speed passenger cars, despite the difference in vehicle speeds, the need for greater protection of truck axles against high-speed failures is indicated. Thus, protection against scoring becomes a recognized factor in lubrication requirements of commercial vehicle axles.

The urgent necessity for improvements in rear-axle lubricants was recognized several years ago by a number of companies engaged in the manufacture of gear lubricants and their additives and some of the advances in the field were discussed recently by Leonard Raymond, of the Socony Mobil Oil Company Inc., before a meeting of the Society of Automotive Engineers Inc.

New products

Several independent projects of extensive scope were undertaken in interested laboratories with the objective of producing new gear lubricants having the anti-scoring advantages of the active-sulphur passenger-car type and the desirable anti-wear and low-friction requirements of the inactive-torque type. The new product is a multi-purpose lubricant and can meet the most critical lubrication requirements of both high-powered passenger cars and commercial vehicles. By comparison, all other lubricants have only a limited field of application. It meets the requirements of passenger-cars with "green" gears, being far superior in anti-weld protection to any lubricant except the single-purpose lead-soap, active sulphur type, and far exceeds all previous truck-type lubricants in protection of heavy-duty truck axles.

The active elements in the new type of multi-purpose gear lubricant are sulphur, chlorine and phosphorous, which act as potent anti-weld materials under high temperature and pressure conditions by chemically forming contaminating surface films which are strong in compression and weak in shear. Through controlled activity they supply the need for the higher chemical activity needed quickly at high sliding velocities and pressures while retaining essential anti-wear and high-torque protection characteristics. The chemical activity is carefully controlled to avoid undesirable corrosion, oxidation and thermal-instability effects. When used in combination with stable base stocks, the result is controlled surface activity for maximum EP effectiveness with adequate thermal and chemical stability and other desirable properties.

In the U.S.A. the testing of the new gear lubricants has involved extensive work in passenger-car and commercial-vehicle axles in the laboratory and on the road by vehicle builders, oil companies, additive manufacturers and the Army. The work has totalled many millions of miles of extremely severe service testing in a variety of equipment and under different operating conditions; in many cases direct comparisons with older types of gear lubricants were made.

The service tests have covered the gamut from short-time high-speed tests in passenger cars to long endurance tests under very severe conditions in both passenger cars and heavy-duty commercial vehicles.

Many of the test vehicles were selected by the vehicle or axle builder because of serious, prohibitive and premature axle failures. Despite the severity of the operations and the unsatisfactory record of the previously recommended lubricants, the new-type products invariably showed a tremendous improvement in performance and elimination of lubrication difficulties. In addition to these special tests, large-scale

area tests were conducted in which the new type of lubricant replaced the older lubricant in a large marketing area on a commercial basis. Testing of this nature is aimed at uncovering any unanticipated flaws, but none were indicated.

This very significant advance in gear and bearing protection by the new super-duty-type lubricant has not been made at the expense of other important lubricating properties. It has excellent resistance against oxidation and deterioration under high-temperature conditions in the presence of catalytic metals. It also has the necessary foam-resistant and storage-stability characteristics; and is compatible with the older types of gear lubricants, although mixing with them is not recommended as it detracts from the superior qualities built into the new product.

All the field tests were conducted in single or double-reduction hypoid axles. Some experience has also been obtained in worm-gear applications. As an example, after successful experience in a West Coast Fleet with a lubricant of 6-Reference level, one company changed to a 10-level product and reports very successful use to date in worm gears. Another company started worm-gear-axle tests several months ago in 70,000 lb dual-worm-gear units running from Denver to Salt Lake City with satisfactory results to date.

Reference gear lubricants

Mention has been made of a series of Reference Gear lubricants used as standards of anti-weld or anti-scoring protection. The Reference Gear Lubricant scale runs from 0 to 15, with 0 (straight mineral oil) being the low end of the scale and 15 (high EP additive content) being the top performance level. This reference quality number is followed by a 75, 80, 90 or 140 (as 15-90) to designate the viscosity grade. The reference quality numbers apply solely to anti-scoring level and cannot be used to describe the performance of a lubricant under high-torque, low-speed conditions. These reference gear lubricants have been used by many of the passenger-car builders to establish the requirement level of their gears both as "green" gears and after a 1,000 mile run-in. Sands[5] has reported that "in some cases the scoring level requirement on run-in axles was in the range of 4-6 Performance Numbers lower than for green axles." The requirement level varied greatly among different axles and with different test conditions.

Dirt, deterioration and gear lubricant change

Even the best of lubricants and designs can be nullified by abrasives. Dirt, particularly when it is abrasive in nature, is one of the worst enemies of a rear axle. Rear axles during their fabrication and assembly tend to accumulate harmful materials from different sources, including sand, metal particles, and flakes of paint. Some of these "built-in" materials can, and do, cause rapid destruction of gears and bearings, and manufacturers have taken various means, including the use of 2,000 lb water pressure in one case, to flush out these materials before final assembly. Experience of at least one manufacturer would indicate that one of the large incidental benefits derived from the phosphate-coating process is that the passage of the parts through the sequence of baths cleans them very thoroughly.

The question of frequency of gear-lubricant change is one which continues to be somewhat controversial. Some passenger-car builders now recommend no gear-lubricant change, while others recommend change and refill either yearly or at 10,000-mile intervals. From the standpoint of the passenger-car owner, it would appear that a very small additional price is paid for what can amount to a large increase in protection when the gear lubricant is changed at 10,000 miles or seasonally, instead of less frequently or not at all. It might be mentioned in this connection that gear-lubricant change is about the lowest item in the total cost of operating a vehicle. The cost of an axle repair is prohibitive in comparison, even without the additional cost and annoyance entailed by a vehicle being out of service.

Future problems

It is probable that axles will have a harder rather than an easier job in the future. In some cases, the axle may be the key to the design of a vehicle. It is encouraging to note the large growth in technical manpower and test facilities assigned to axle research and development by a number of major car and commercial vehicle builders in recent years. This expansion in experimental facilities has included full-scale axle installations in the laboratory and increased road and proving ground tests.

In addition, manufacturers are faced with some problems which may require basic research for their solution. In this connection, more work is being directed along theoretical and mathematical lines to analyse gear stresses under dynamic conditions. Due to the unsteady nature of the power plant and the axle load, the dynamic loads on the gear teeth can be very much greater than those computed on a constant torque or load basis. The Armour Research Foundation is now conducting an excellent programme of determining instantaneous and peak gear loads using special methods and instrumentation. For research on wear and its prevention, the use of radioactive techniques is a very promising tool.

Temperature level is an extremely important factor in axle performance. While the builder realizes and states that bulk-oil temperatures in excess of 250 deg F are detrimental to gears, bearings and lubricant, apparently the temperature load is becoming greater than desirable with current products. A real effort to overcome this difficulty either through design, metallurgy or lubrication, may be necessary. The manufacturer may require new gear steels having higher hot strength and less subject to such things as surface decarburization and annealing at temperatures above 250 deg F. The lubricant supplier may find a way to combine excellent anti-weld activity with even better thermal and chemical stability at high temperatures.

Higher temperatures are also focusing attention on seals for axles and other components, with indications that seals may be the most critical elements in the entire automotive drive line. Higher temperatures lend emphasis to the need for adequate compatibility of seals and petroleum products. The increasing need for co-operative action among the vehicle builders, the seal manufacturers, and the lubricant suppliers is being met by SAE, ASTM and CRC groups.

The increasing recognition in various quarters of the value of closer co-operation between equipment manufacturers and lubricant suppliers is a promising factor for the future. One such possible combination which will pose a tough problem is an automatic transmission-hypoid axle unit utilizing a common lubricant. There is need for basic research on mechanisms of gear-surface protection, including information on the change in surface condition and chemical activity level during and after break-in, and the cause and effect of higher temperatures.

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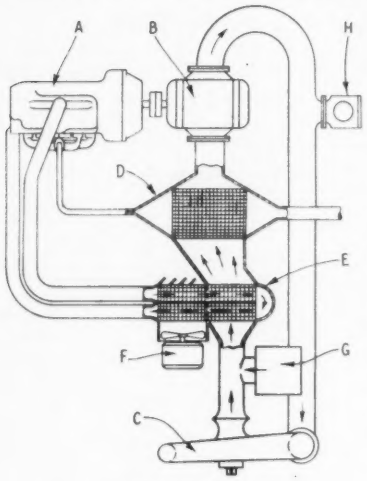
CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Pneumatic transmission system

A hydrostatic transmission, in which an engine-driven pump feeds a hydraulic motor or turbine delivering the output power, operates as a torque converter and provides a flexible and stepless drive. It has the disadvantage, however, of a relatively low overall efficiency. The closed-circuit, recuperatively heated, pneumatic transmission, which is the subject of this invention, is claimed to avoid this disadvantage and to be applicable to a motor vehicle.

In the schematic layout, a conventional piston engine A drives a Roots type compressor B, either directly, as shown, or



No. 763433

through an increasing gear. The delivery side of the compressor is connected by ducting to the entry of an inward-flow turbine C and the turbine exhausts back to the suction side of the compressor. Heat remaining in the air after expansion through the turbine is, therefore, not wasted.

The engine exhaust gases are passed through a heat exchanger D, where more heat is acquired by the throughput air, and the efficiency of the system is further improved. The gases are discharged to atmosphere. Additionally, the engine coolant may furnish a source of heat by means of a second exchanger E located at a point before the exhaust gas unit D. The temperature of the coolant may be further lowered by air cooling by means of a fan F before being returned to the engine. An inward-flow air valve G, equipped with an air filter, is fitted in the return ducting between the turbine and the compressor to provide automatic compensation of any air losses in the system.

Alternatively, exhaust heat may be utilized by injecting the exhaust gases directly into the working medium circuit. In such a system, a blow-off valve H will be provided.

Power from the turbine shaft to the road wheels will be transmitted through a reducing gear and a differential mechanism. The need for a differential gear could be obviated by the provision of separate drives from a pair of coaxial turbines. *Patent No.* 763433. *Sir George Godfrey and Partners (Holdings), Ltd.*

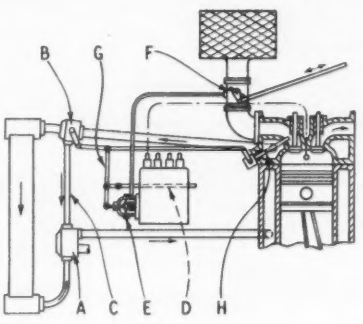
Regulation of petrol-injection engines

With the primary object of facilitating starting, the fuel:air ratio of this engine is controlled by a single thermostat. This thermostat is influenced, either directly or indirectly, by the cooling water, the cooling air, or the engine lubricant, and may be arranged to regulate also the temperature of the medium that acts upon it. When an engine is cold, part of the fuel injected may be deposited on the cylinder walls and the fuel:air ratio of the mixture may be lower than is desirable. Thermostatic regulation ensures that an increased quantity of fuel is injected at such times so that a favourable mixture is available for ignition.

In the example illustrated, the cooling water is circulated through the jackets of the cylinders and the cylinder heads by a pump A and returns past a thermostatic valve B proportioning the flow between the radiator and a by-pass C in the usual manner. Quantitative regulation of the fuel delivered by injection pump unit is by a rack D under the influence of a suction diaphragm E displaced by the depression in the air intake manifold at a point adjacent to the controlling throttle valve F. A lever G connected to the diaphragm E is linked to the rack D and to the thermostatic valve B. Additional regulation in dependence upon the temperature of the combustion air is also provided. A thermostat H projecting through the cylinder head jacket to the air inlet passage is linked to the upper end of lever G.

In operation, the pump rack is moved in dependence on the manifold depression and thus in conformity with the amount of air drawn into the engine cylinders. Temperature-dependent regulation is superimposed. When the cooling water is relatively cold the thermostatic valve B will move G to increase the amount of fuel injected. Thermostat H, responsive to the temperature of the cooling water and also to the temperature of the air for combustion, is arranged to reduce the amount of fuel injected as the temperatures of these elements increase.

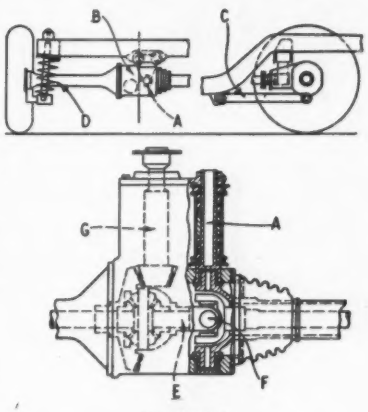
It will be noted that the action of thermostat H will influence the thermostatic valve B. In suitable instances, therefore, the thermostatic element in valve B may be dispensed with and the system controlled by a single thermostat H. *Patent No.* 761094. *Daimler-Benz A.G. (Germany).*



No. 761094

Arrangement of axle gearing

In an independent rear suspension the final drive and the differential gearing are mounted on one of the rocking half-axles. The arrangement provides a simple construction requiring a single articulation point only, dispensing with a separate mounting of the gearing unit on the vehicle structure, and increasing the rocking length of the half-axles and thus reducing camber variation. Weight distribution is scarcely affected as the propeller



No. 761942

shaft lies closely adjacent to the longitudinal central plane of the vehicle.

At their inner ends the half-axles are articulated about pivot pin A, on the off side directly, and on the near side through the attached or integral gear housing B. Pin A is carried by a bracket secured by two rubber-bushed pins to a transverse member of the vehicle structure. The outer ends of the half-axles are guided by links C and sprung by helical springs D assisted by co-axially arranged telescopic shock absorbers.

In the housing B are mounted the bevel pinion and crown wheel and the differential gearing which drives the near-side axle shaft and a short shaft E that is connected through a universal joint F to the off-side axle shaft. The pivot centre of the universal joint is on the axis of pin A. Pinion shaft G and pivot pin A are disposed as close to each other as is possible and spaced on either side of the longitudinal central plane of the vehicle so that the two axle shafts are of the same length and interchangeable.

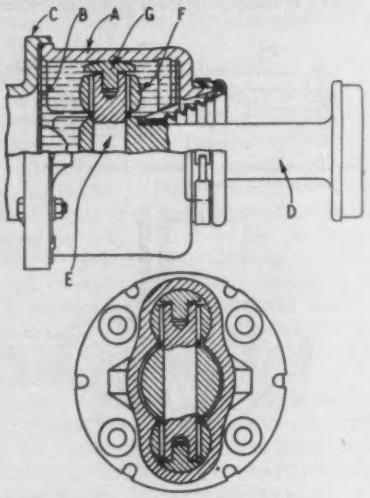
Housing B is attached to the ends of pin A, which is mounted in two rubber bushes in the suspension bracket, and the forked end of the off-side half-axle is trunnioned in the rear end of the pin and a coaxial bushing in the housing. *Patent No.* 761942. *Daimler-Benz A.G. (Germany).*

Sliding universal joint

This two-trunnion type universal joint has its sealing boot so constructed and arranged that the common disadvantages of distension in operation and exposure to matter thrown up by the road wheels are obviated. Since oil cannot collect to one side of the boot, the joint will not become unbalanced and cause undesirable vibration.

The hollow body A, adapted to contain lubricant, is sealed at one end by a cover plate B and secured to a companion flange C. At the other end is an opening, defined by a neck, through which the end of the propeller shaft D projects into the body. In the end of shaft D is a cross-pin E, having a ball F journalled on needle rollers on each end. The balls are engaged in cylindrical guides formed in the body and the assembly is centred by buttons G, mounted in the ends of pin E with an interposed spring washer.

Formed wholly of rubber-like material, the corrugated boot of frusto-conical shape is clamped by a metal band at its



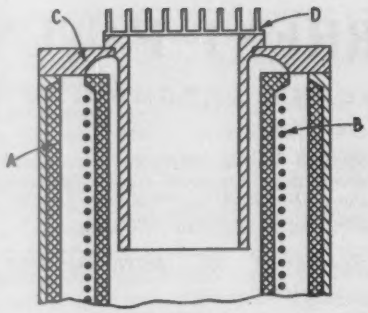
No. 763307

smaller end to the shaft D and at its larger end is turned over the neck on the body and secured by a second metal clamp. *Patent No.* 763307. *Chrysler Corporation (U.S.A.)*

Differentially hardened cylinder liners

If cast iron cylinder liners are hardened and tempered to change the microstructure from pearlitic to martensitic the strength of the material is reduced from that obtaining in the "as cast" condition. As a consequence, particularly in the case of wet-type liners suspended in the cylinder block by an upper flange, the possibility arises of the liner breaking away from the flange. Accordingly, it is proposed to harden the body of the liner selectively and leave the flange in a condition which, at least, does not impair the "as cast" strength.

The liner is placed in a furnace, comprising refractory linings A and a heating element B, with its flange supported on a plate C and protruding from the furnace. In the furnace the body of the liner is heated to a temperature of 885-890 deg C, but the flange is not raised to the change point, approximately 760 deg C. If necessary, a ribbed cap D of copper may be



No. 763492

placed in contact with the flange and a current of air directed over it to dissipate heat. When the liner is quenched in oil and tempered at 370-375 deg C, the main part will have a martensitic structure and the flange an improved pearlitic structure.

Tests have shown that a fully hardened (martensitic) flange broke under Izod impact of 90-100 in-lb, an "as cast" flange at 260-290 in-lb, and the flange of a liner heat-treated according to the invention of 290-320 in-lb.

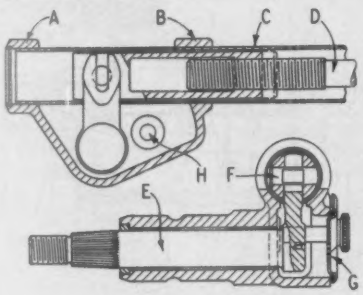
The point at which the change from martensitic to pearlitic structure is to occur is determined in relation to the travel of the top ring of the piston to be used. If the top ring is so high that it would enter the unhardened zone, a short length of the upper end of the liner is chromium plated in order to ensure that no undue wear will occur. *Patent No.* 763492. *Wellworthy Ltd.*

Steering gear

By its simple design and construction, this steering gear is easy and inexpensive to manufacture. The housing is formed with opposed annular parts A and B which receive one end of tubular member C which encloses the steering shaft D for its entire length. Rocker shaft E is journalled in an extension of the housing and the bifurcated end of the rocker arm extends through a slot formed in the tubular member C.

The inner surface of member C adjacent the housing is machined to constitute a bearing surface for a tubular nut engaged on the threaded end of steering shaft D. At its remote end, the nut is slotted to accommodate the end of the rocker arm. A cross pin F journalled in the nut is provided with flats to slidably engage the slot in the end of the rocker arm.

A cover plate G closing the housing is secured by a milled-head screw engaged in tapped hole H, and the screw shank constitutes a stop limiting the angular movement of the rocker arm. *Patent No.* 763302. *R. Bishop, R. H. Johnston, and G. B. R. Whitlock.*

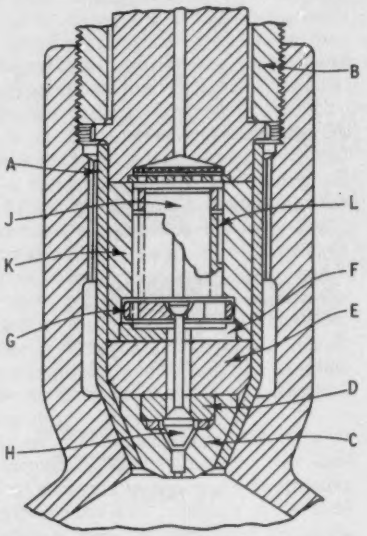


No. 763302

Magnetically closed fuel injector

In the orthodox fuel injector the needle valve is of the inwardly opening type and is held closed by a helical compression spring. At some operating speeds much of the pressure energy supplied by the injection pump is used to hold open the needle against increasing spring pressure instead of being available to atomize the fuel as it is sprayed into the combustion space. As wear occurs in the assembly the operative length of the spring is increased and the decreased spring pressure leads to premature injection of fuel and possibly to a later and less positive cut-off on closure.

The invention provides an outwardly opening valve held in the closed position by a permanent magnet. Since the flux density at the poles of the magnet decreases rapidly as the distance of the armature from the magnet increases, the energy required to hold the valve open is less than that necessary to open the valve. When wear occurs between the valve and the seating,



No. 763799

the air gap is decreased and consequently the opening pressure will be increased. Such an increase is preferable to the converse which applies in the case of the orthodox needle valve.

A cylindrical body A contains the valve, magnet, and nozzle assembly and is seated in the cylinder head or precombustion chamber and secured by a fuel adaptor drawn down by a sleeve nut B. The nozzle tip C embodying the discharge orifice is fitted in the coned end of the body and supports the valve seat D on a spacer ring of a relatively soft metal. Abutting the nozzle tip and the valve seat is a spacer E carrying a recessed and slotted stop plate F. The armature disc G has a clearance hole joined by a slot to the central part-spherical socket from which the valve H is suspended.

For the magnetic circuit a cylindrical magnet J is surrounded by a soft ferromagnetic shell K, in contact with one pole of the magnet and elsewhere spaced from it by a non-magnetic sleeve L. A fine-mesh screen is supported on a perforated disc in the end of the fuel adaptor and any minute ferrous metal particles passing the screen will be attracted to and held by the magnet. *Patent No.* 763799. *Caterpillar Tractor Co. (U.S.A.).*

PATENT RIGHTS FOR DISPOSAL

Independent suspension with front wheel drive

THIS SYSTEM (shown under licence on the gas-turbine Rover car at Earls Court Motor Show 1956), utilizes the cardan shafts as links in the suspension wishbone linkage, and eliminates the troublesome sliding joints in the universals. It also simplifies the general layout, especially when ball-joint suspension is used. The patent rights are sub-divided as follows:

BRITISH PATENT No. 702,454

For all cars other than those having piston-type internal combustion engines of capacity 1000 c.c. or less; an exclusive licence in this category has been acquired by

THE ROVER COMPANY LIMITED.

For cars having piston-type internal combustion engines of 1000 c.c. or less capacity;

THESE RIGHTS ARE FOR DISPOSAL

In addition, there are applications, (Figs. 13, 14 and 15) in both capacity categories, to rear wheel drive with outboard reduction gear; these are expressly suitable for transmitting heavy torques, where the elimination of the sliding joint is of particular value.

FRENCH PATENT No. 1043096

The owners wish to make arrangements in respect of this, with or without division into categories as above, but subject to freedom for the sale or use in France of Rover cars made in Britain under the patent, and to the sale or use in Britain of cars made under the French patent.

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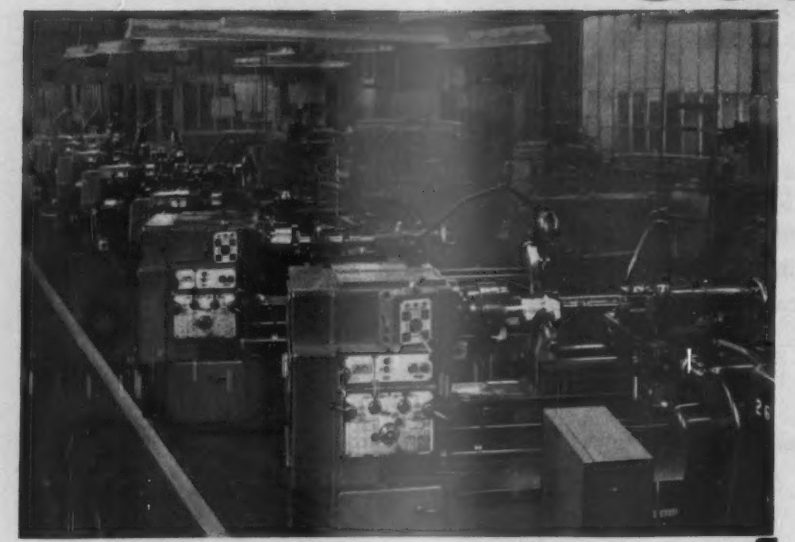

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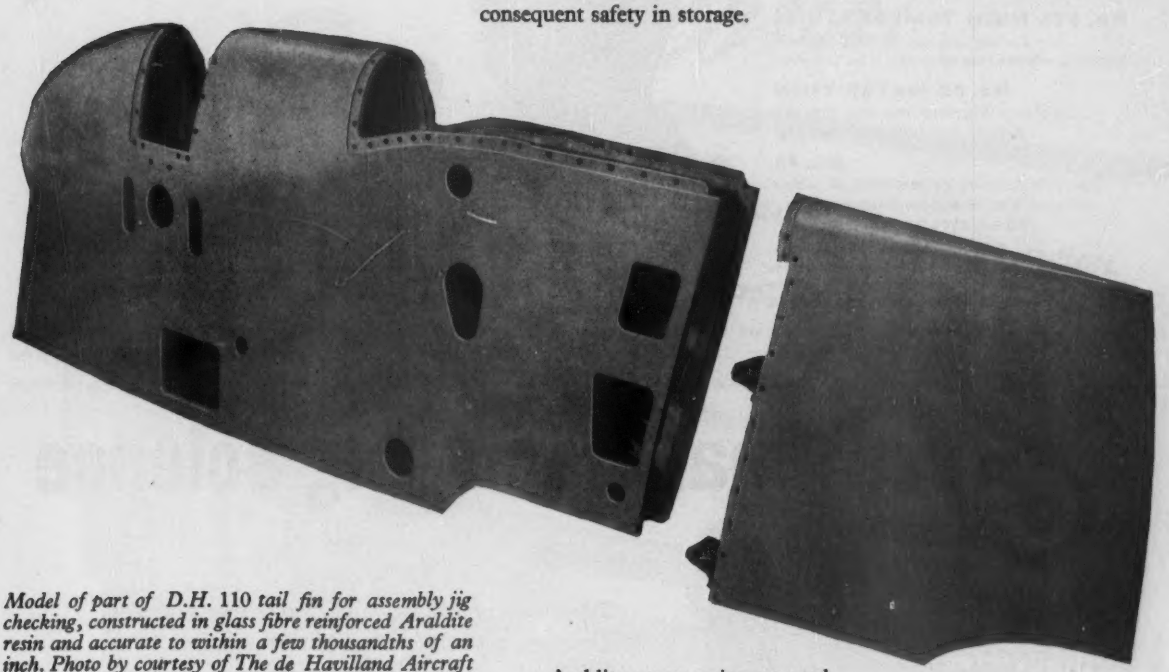
Tooling techniques with Araldite resins are being adopted more and more widely. Models, jigs, fixtures, patterns and metal-forming tools, including hammer forms, stretch blocks, rubber press tools, drop-hammer and draw dies, can all be made by simple methods with virtually no machining and a minimum of skilled manual work. Araldite tooling resins are used as gravity cast mineral-filled mixtures or with glass fibre mat or woven reinforcement, the latter providing strong, dimensionally stable, but lightweight surfaces and structures. Advantages of Araldite in tool-making include ease and speed of production, low production costs, light weight and ease of handling, negligible shrinkage on curing, accuracy of reproduction, dimensional stability, durability, resistance to cutting-oils and die lubricants, resistance to moisture and chemical attack, with consequent safety in storage.



Model of part of D.H. 110 tail fin for assembly jig checking, constructed in glass fibre reinforced Araldite resin and accurate to within a few thousandths of an inch. Photo by courtesy of The de Havilland Aircraft Co. Ltd.

The recommended techniques are simple and easily acquired. Full details and practical assistance are available upon request. May we send you a copy of our new publication on Araldite for tools, jigs and fixtures?

Araldite epoxy resins are used

- **for bonding metals, porcelain, glass, etc.**
- **for casting high grade solid electrical insulation**
- **for impregnating, potting or sealing electrical windings and components**
- **for producing glass fibre laminates**
- **for making patterns, models, jigs and tools**
- **as fillers for sheet metal work**
- **as protective coatings for metal, wood and ceramic surfaces**

Araldite *epoxy resins*

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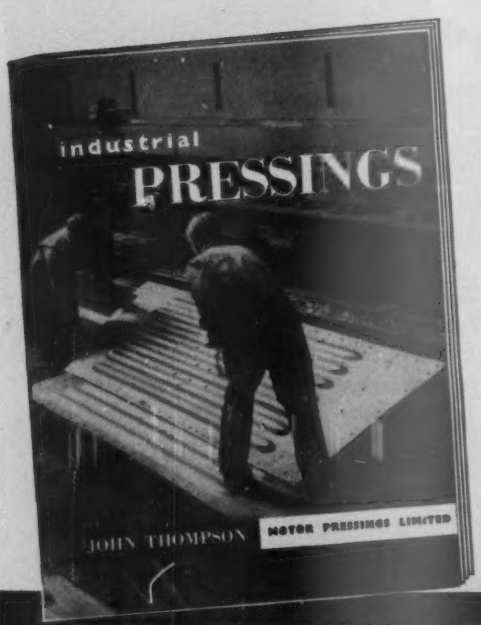

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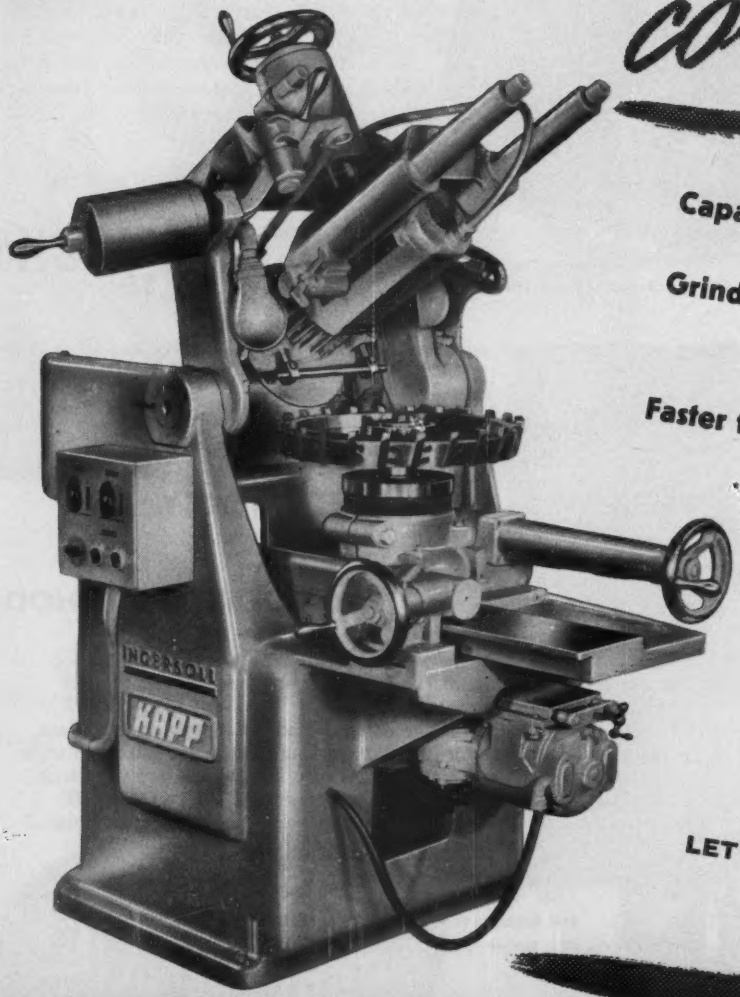

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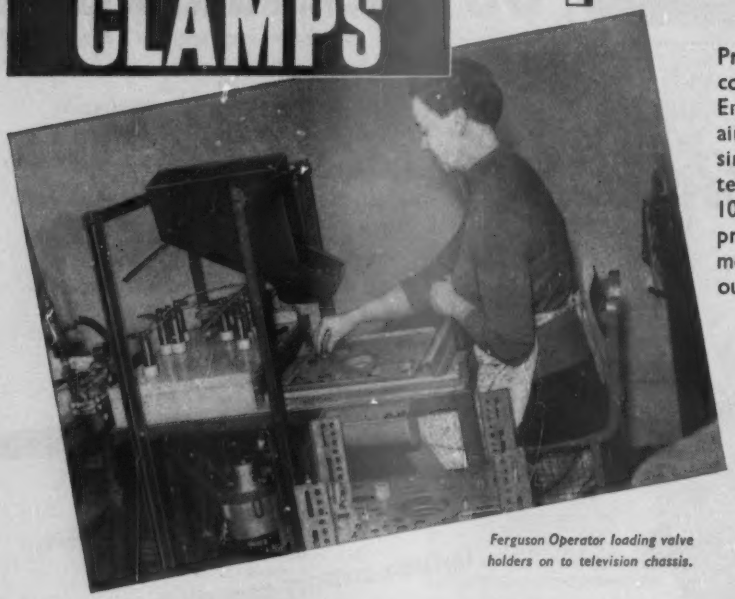

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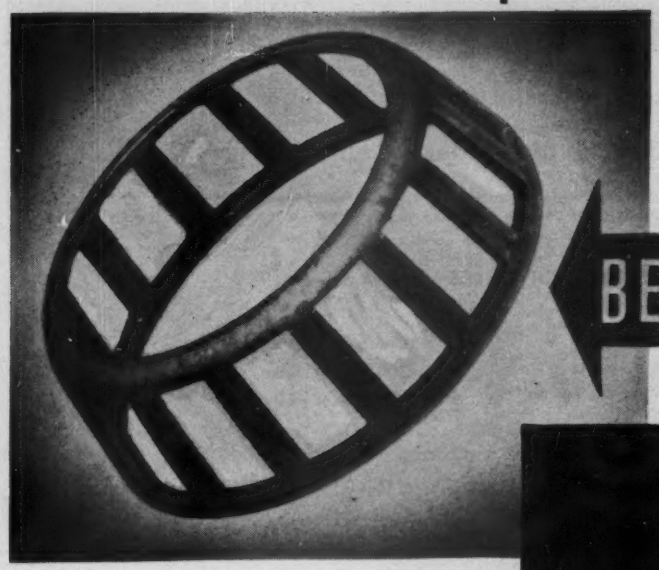

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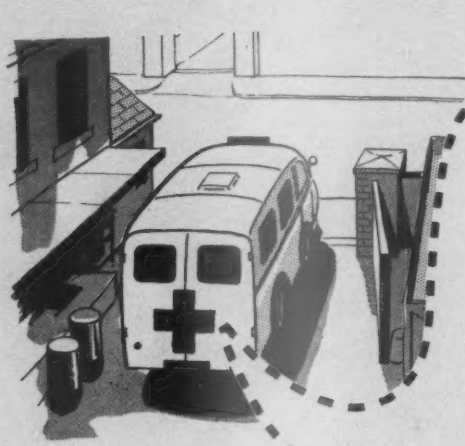

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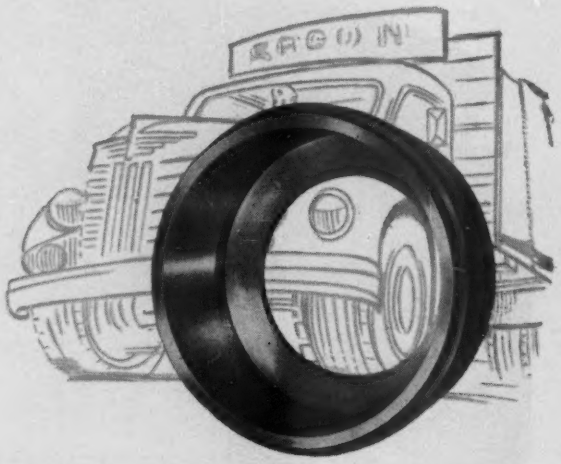

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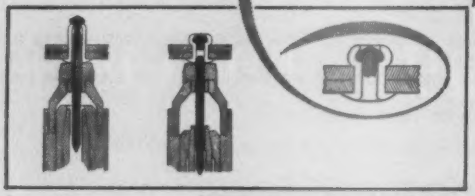

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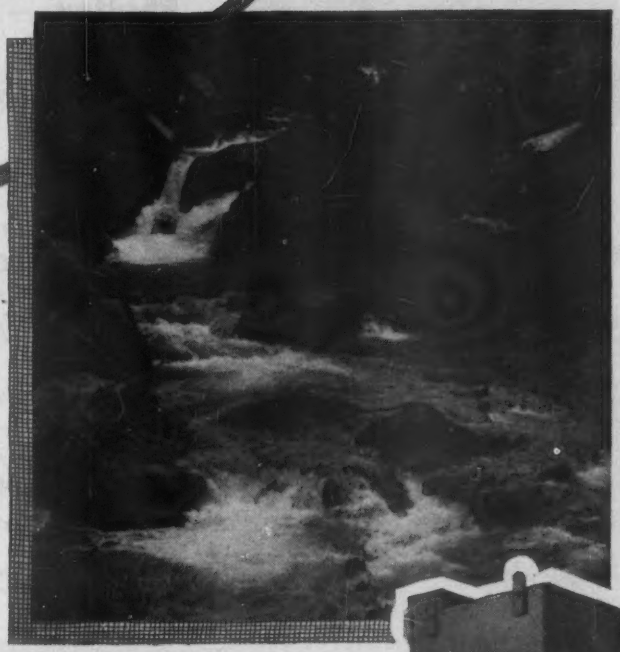

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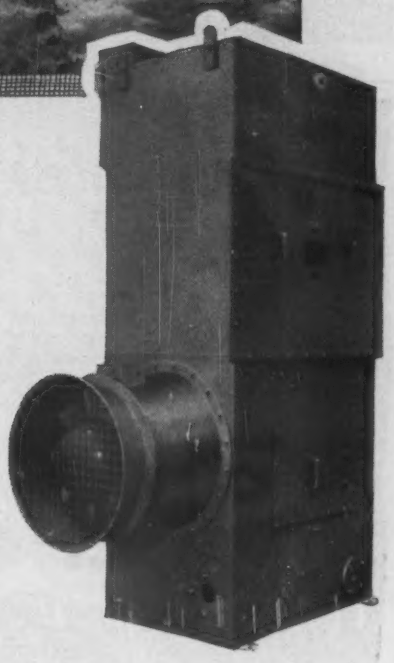

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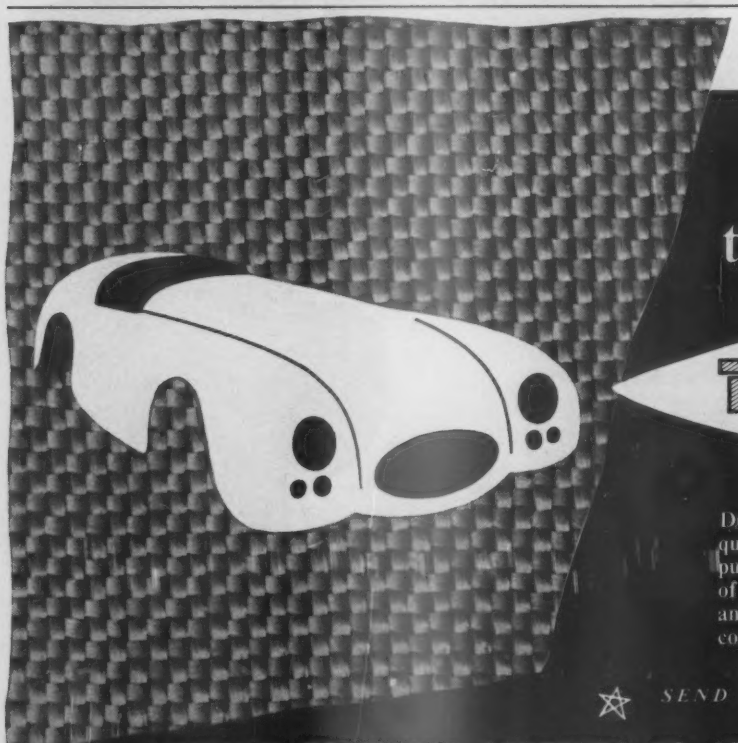

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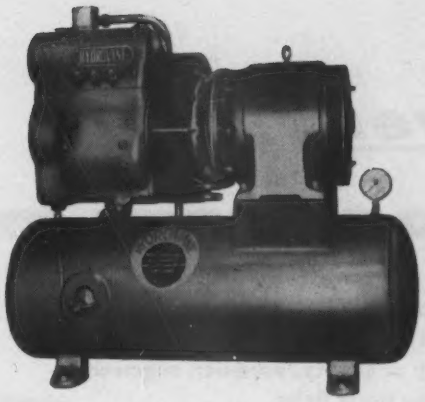

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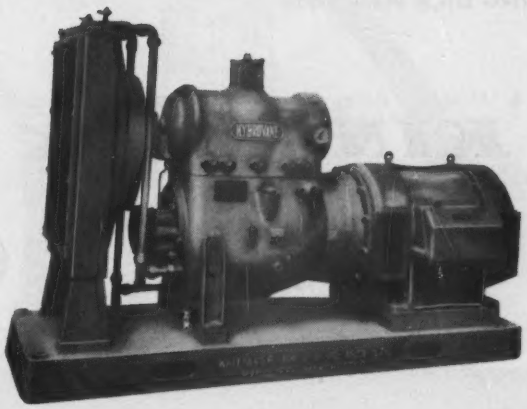

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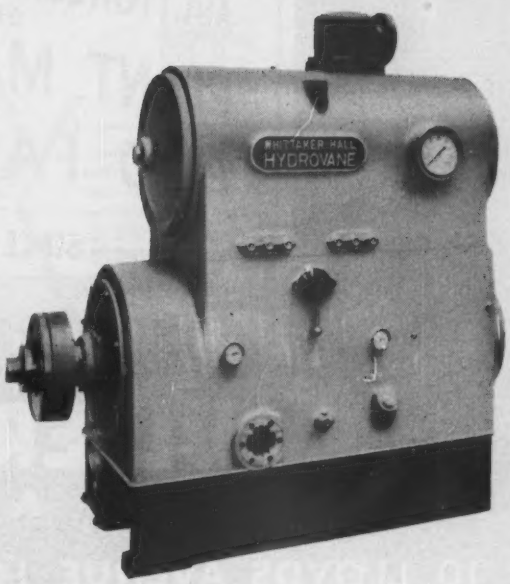

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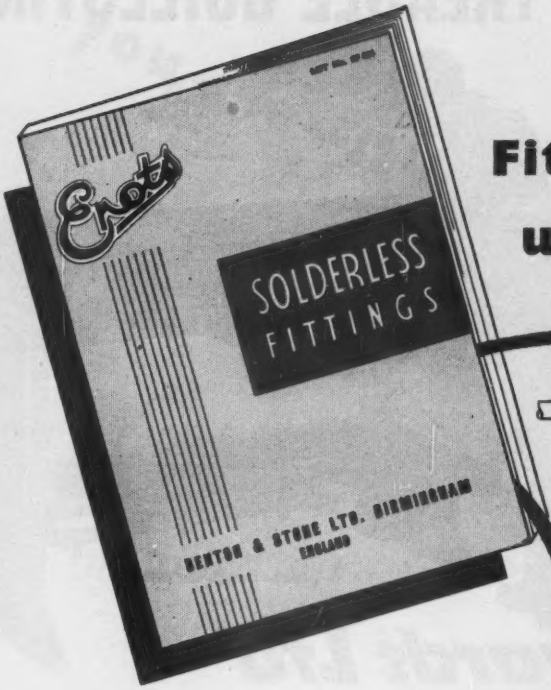
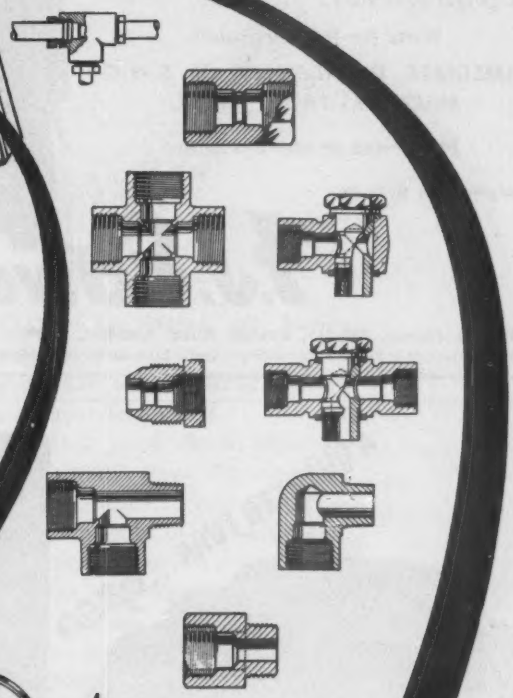


Fitted any good unions lately?



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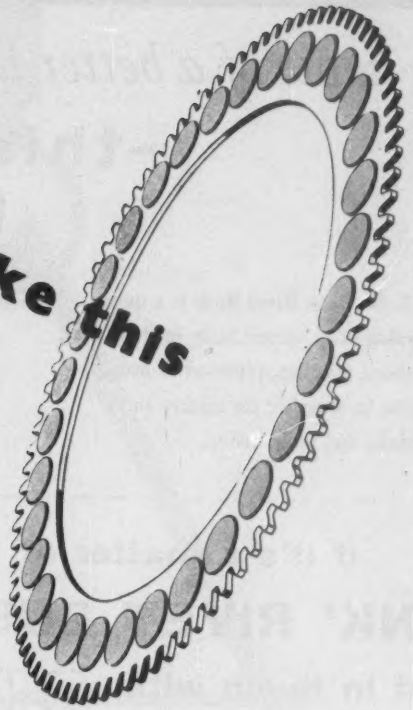

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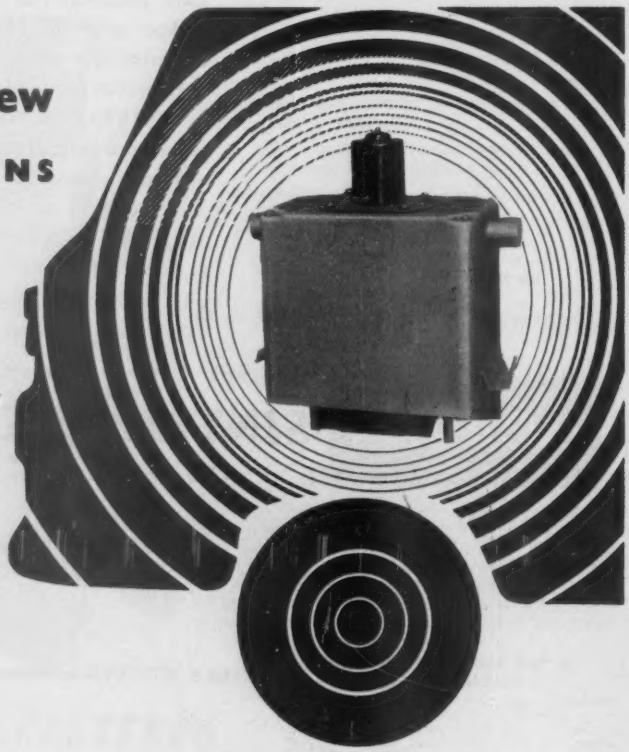

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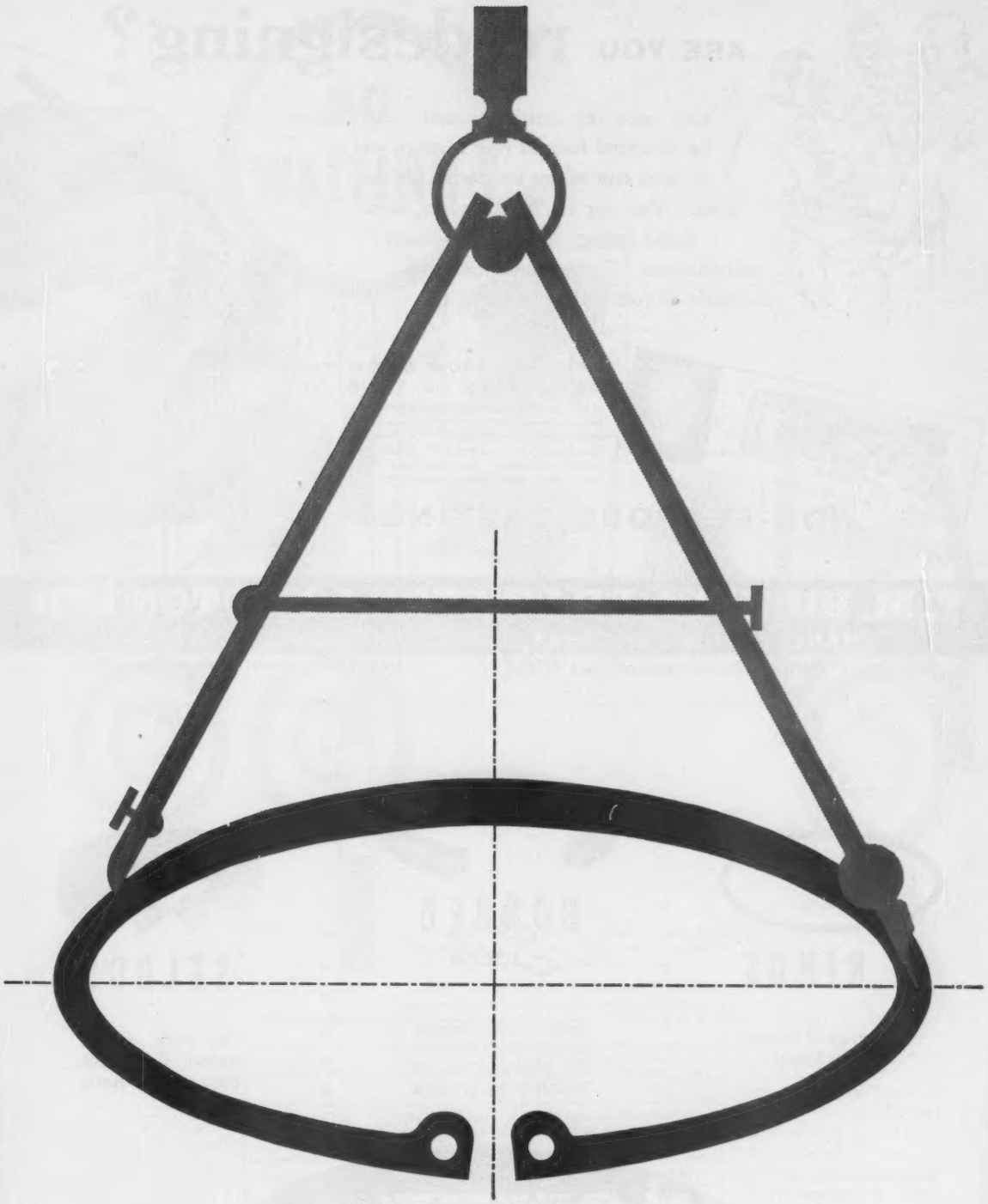

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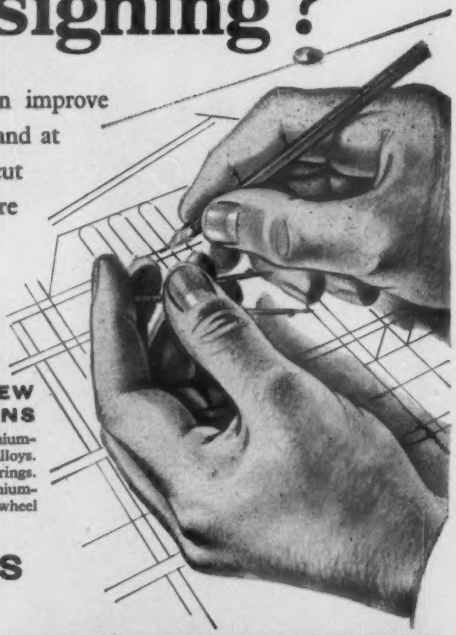

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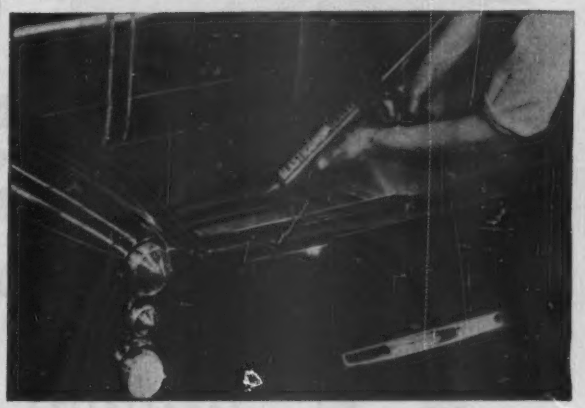

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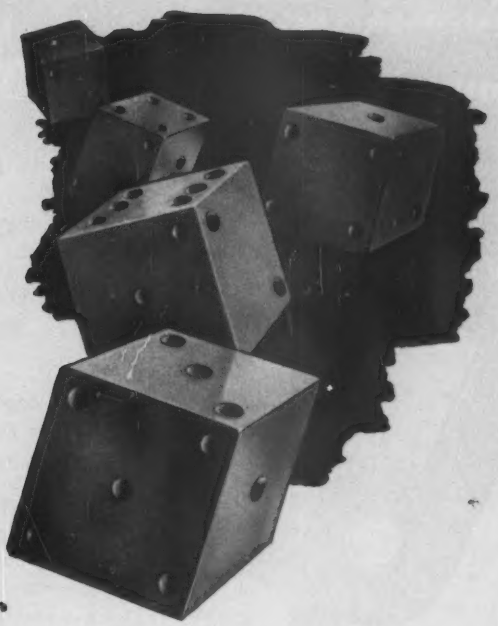

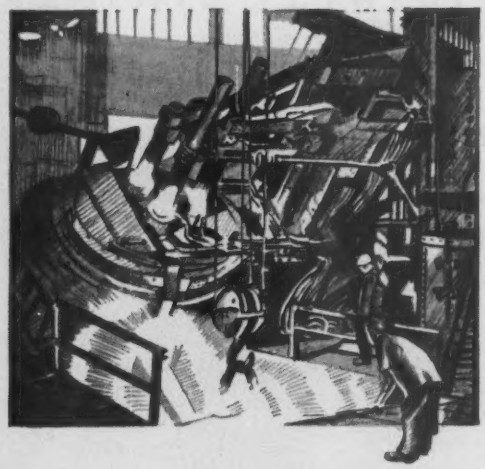

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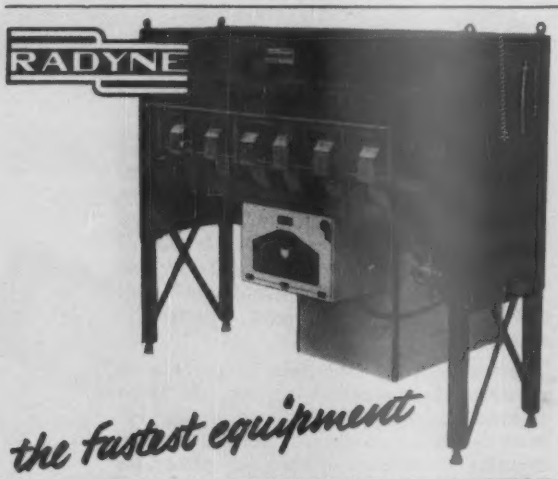

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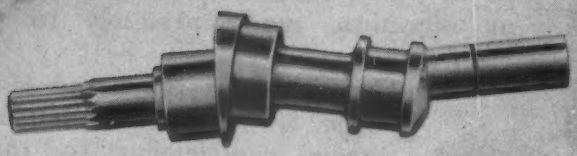

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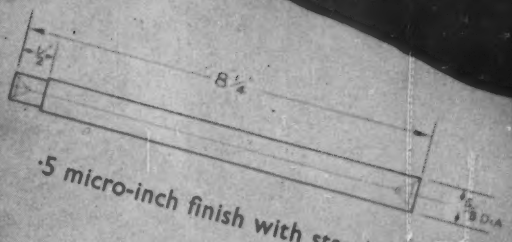

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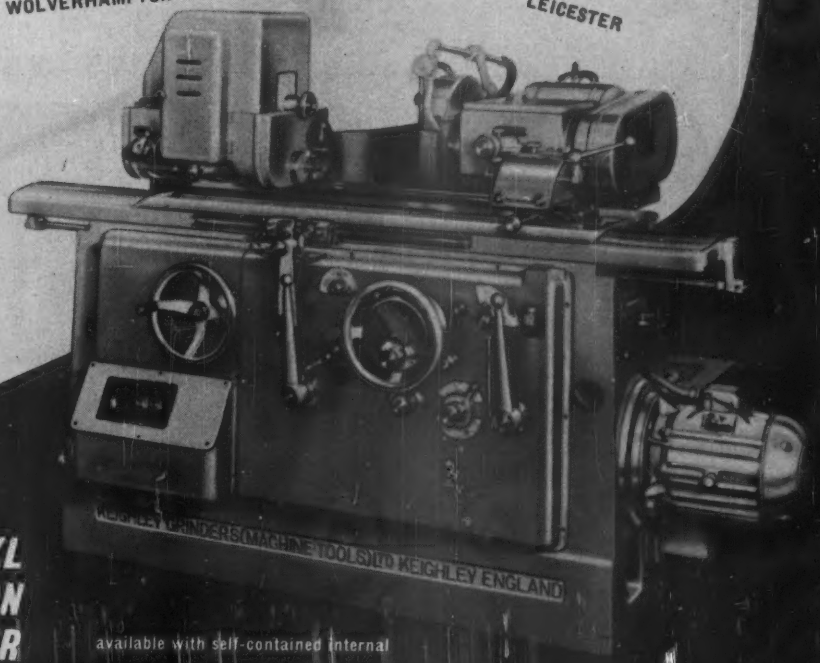

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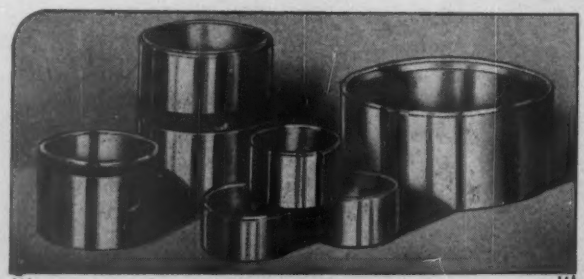

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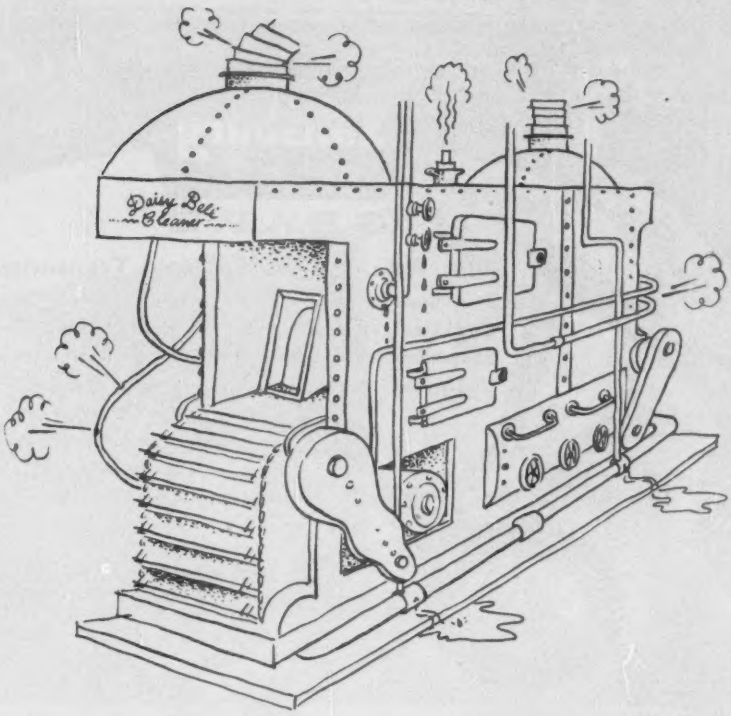

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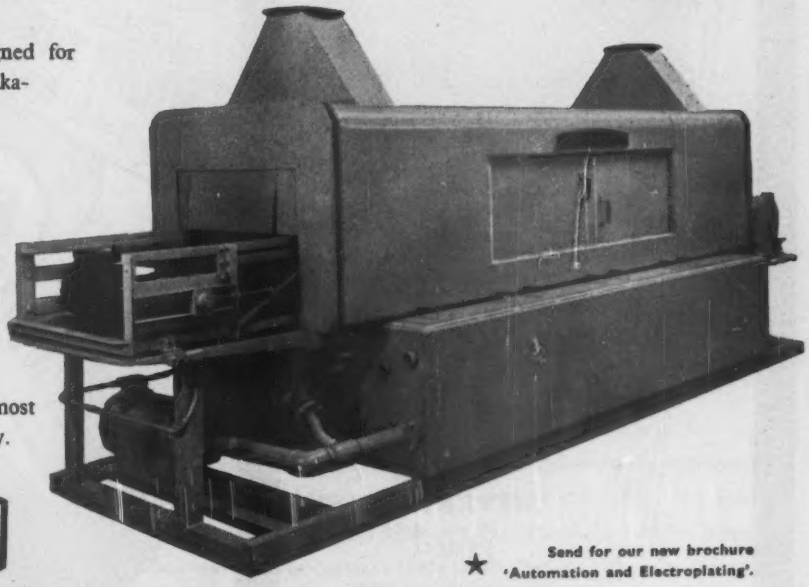

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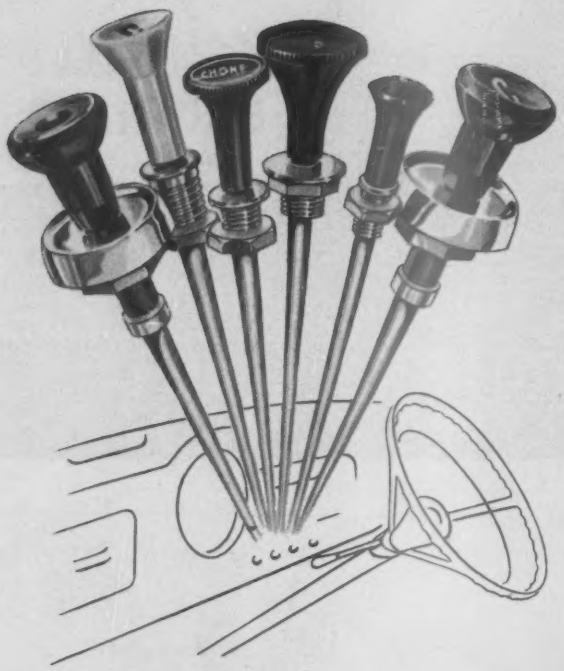

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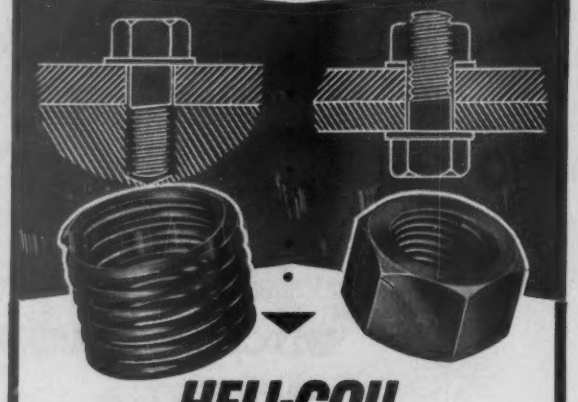

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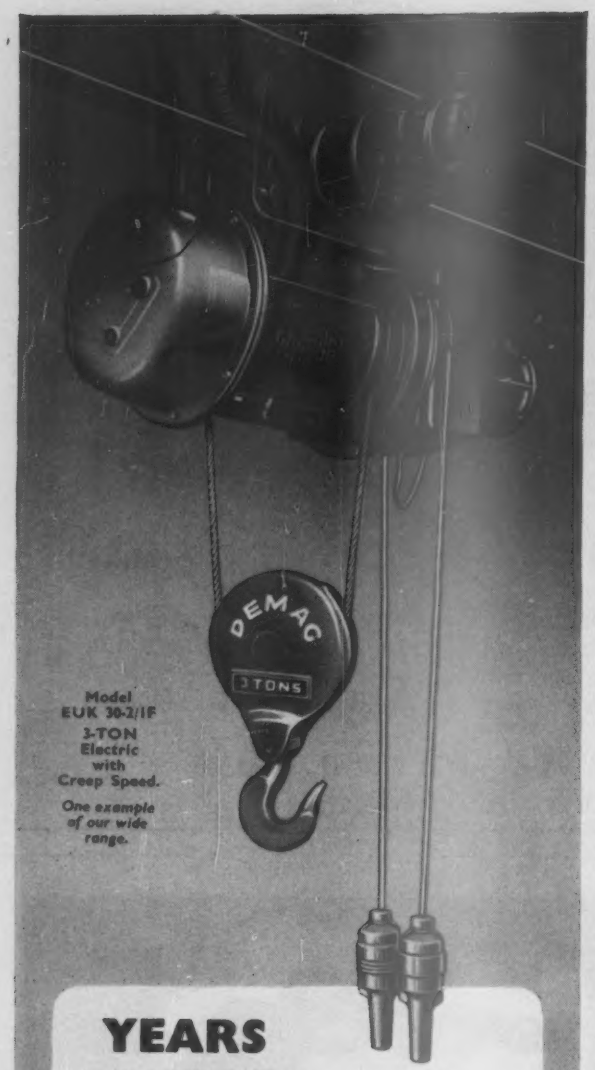

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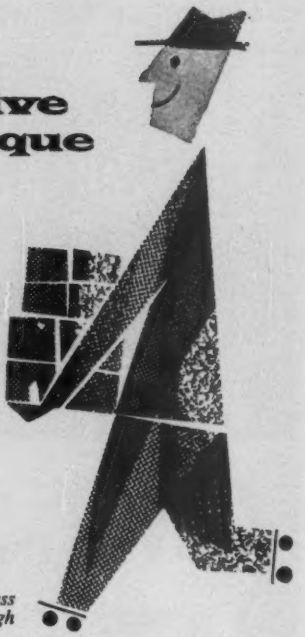

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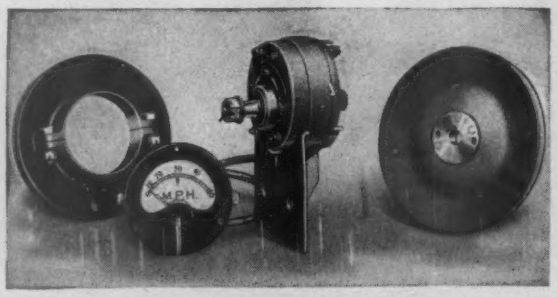

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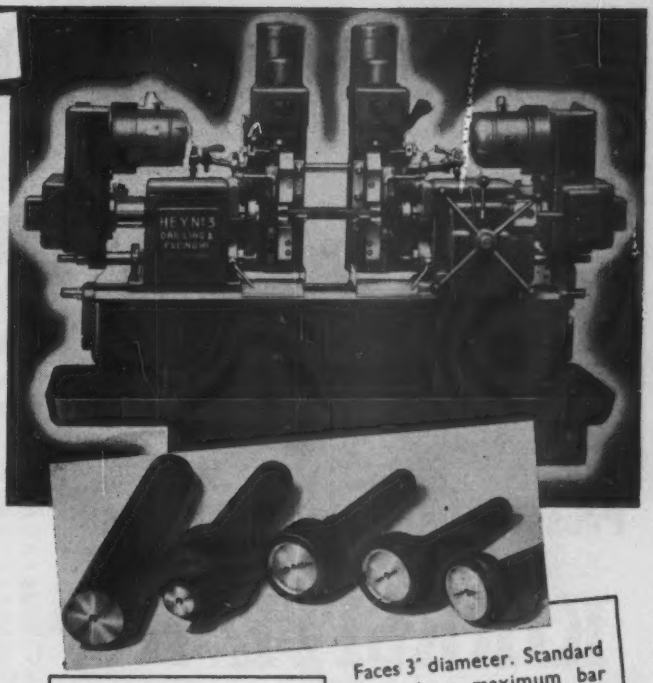

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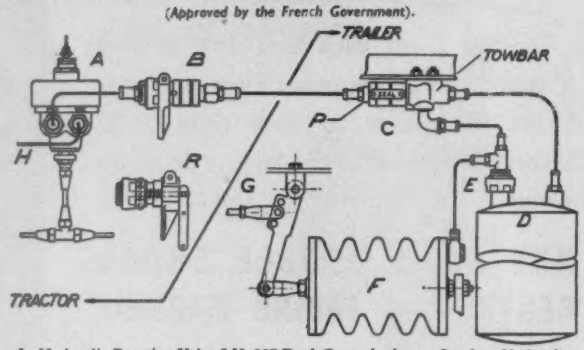

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B. "C" type Coupling with Auto. Shut-off Valve, attached to rear of Tractor.
C. Breakaway Valve SBS. 256 (or SBS. 257) attached to Tow Bar of Trailer.
D. Vacuum Tank on Trailer.
E. Non-return Valve.
F. Power Unit on Trailer operating Brakes.
G. Trailer Brake linkage.
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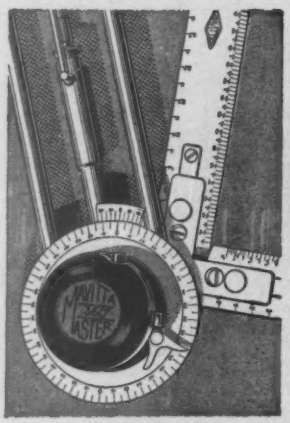

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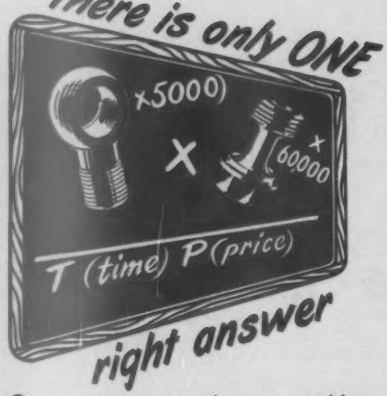

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NOTICE is hereby given that Roy Selden Sanford and others seek leave to amend the Complete Specification of the Application for Letters Patent No. 775,052 for an invention entitled "Cooling Arrangement for Brake Mechanism."

PARTICULARS of the proposed amendments were set forth in the Official Journal (Patents), No. 3,582, dated October 9th, 1957.

ANY person may give Notice of Opposition to the amendment by leaving Patents Form No. 36 at the Patent Office, 25 Southampton Buildings, London, W.C.2, on or before November 9th, 1957.

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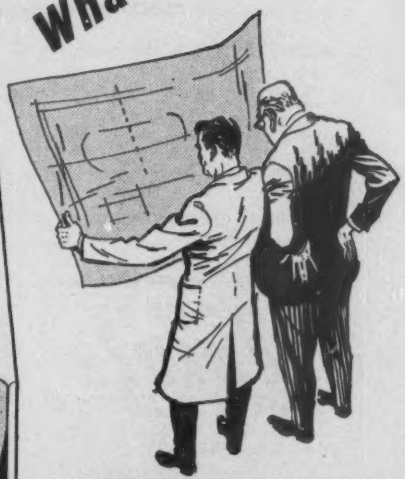

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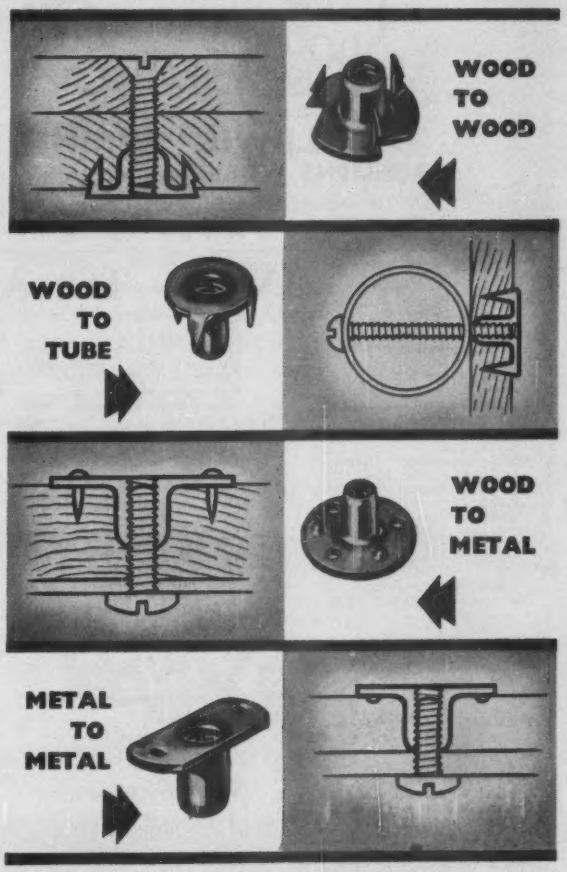

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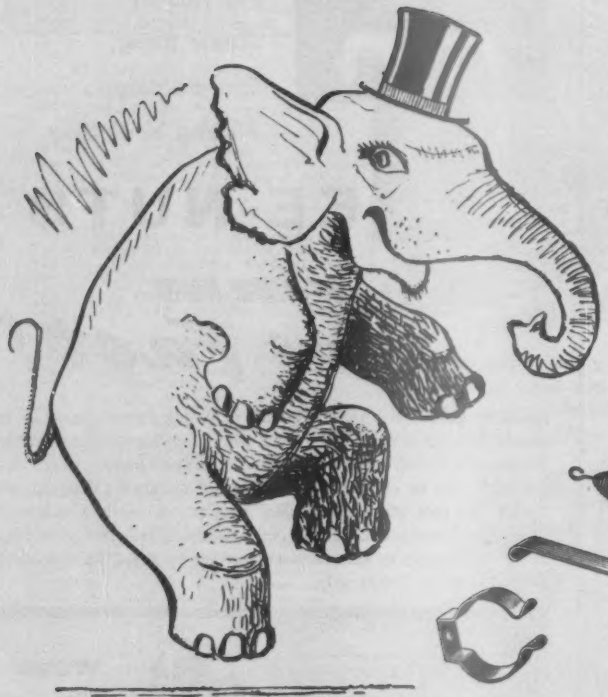

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INDEX TO ADVERTISEMENTS

	PAGE
Abingdon King Dick Ltd.	90
AC-Delco Division of General Motors Ltd.	20
Acheson Colloids Ltd.	16
Acrow (Engineers) Ltd.	112
Adamant Engineering Co. Ltd.	56
Aero Research Ltd.	79
Aircraft - Marine Products (G.B.) Ltd.	24
Almco Supersheen Division of Great Britain Ltd.	83
Aluminium Bronze Co. Ltd.	111
Amal Ltd.	108
Archdale, James, & Co. Ltd.	22
Armstrong Patents Ltd.	110
Armstrong Whitworth & Co. (Pneumatic Tools) Ltd.	36
Atlas Copco AB	39
Automotive Engineering Ltd.	95
Automotive Products Co. Ltd.	5, 6, 7, 8
Benton & Stone Ltd.	91
Berry, Richard, & Son	98
Birkett, T. M., Billington & Newton Ltd.	96
Borg - Warner Ltd. (Morse Chain Div.)	42
British Belting & Asbestos Ltd.	Cover ii
British Electrical Development Association	48
British Geon Ltd.	4
British Oxygen Co. Ltd., The	30
British Petroleum Co. Ltd., The	2
British Piston Ring Co. Ltd., The	26
British Thomson-Houston Co. Ltd.	112
Broom & Wade Ltd.	50
Brown, David, Corporation (Sales) Ltd., The	1, 104
Brown, N. C.	116
B.S.A. Tools Ltd.	Cover iv
Bull, John, Rubber Co. Ltd.	115
Burton Griffiths & Co. Ltd.	Cover iv
Cape Asbestos Co. Ltd., The	70, 71
Carr Fastener Co. Ltd., The	117
Cashmore, John, Ltd.	100
C.A.V. Ltd.	72
Charles (Wednesbury) Ltd.	78
Clancey, G., Ltd.	108
Clayton Dewandre Co. Ltd.	31
Clayton-Wright, Howard, Ltd.	55
Collier & Collier Ltd.	106
Concentric Mfg. Co. Ltd.	107
Consolidated Pneumatic Tool Co. Ltd	Cover iii
Cooper & Co. (Birmingham) Ltd.	82
Coopers Mechanical Joints Ltd.	102
Courtaulds Ltd.	68
Coventry Motor Fittings Co. Ltd.	25
Cross Mfg. Co. (1938) Ltd.	104
Dartmouth Auto Castings Ltd.	74
Delaney Gallay Ltd.	94
Desoutter Bros. Ltd.	9
Dowty Seals Ltd.	96
Drummond - Asquith (Sales) Ltd.	58
Dzus Fastener Europe Ltd.	92
Eagle Pencil Co.	117
Eaton Axles Ltd.	67
Edwards, F. J. Ltd.	92
Electro-Chemical Engineering Co. Ltd.	105
Feeny & Johnson Ltd.	114
Feltham, Walter H., & Sons Ltd.	116
Ferodo Ltd.	32, 33
Firth-Vickers Stainless Steels Ltd.	21
Fodens Ltd.	100
Fry's Metal Foundries Ltd.	88
Fuller, Horsey, Sons & Cassell	90
Funditor Ltd.	85
General Electric Co. Ltd., The	60
Gill's Cables Ltd.	110
Girling Ltd.	35
Gordon & Gotch Ltd.	80
Gotham Co. Ltd., The	43
Guest, Keen & Nettlefolds (Midlands) Ltd.	94
Harper, John, & Co. Ltd.	44
Heenan & Froude Ltd.	87
Herbert, Alfred, Ltd.	11, 63
Hey Engineering Co. Ltd.	113
Hoffmann Manufacturing Co. Ltd., The	13
Holroyd, John, & Co. Ltd.	97
Hot Pressed Products	111
Hoyt Metal Co. of Great Britain Ltd., The	114
I.C.I. (General Chemicals, Heat Treatment) Ltd.	15
I.C.I. (Metals) Ltd.	46, 62
I.C.I. (Isocyanates) Ltd.	59
I.C.I. (Marston Excelsior) Ltd.	113
Ilford Ltd.	37
"Independent"	77
Jackson, H., Ltd.	23
Kelseal Ltd.	99
Kirkstall Forge Eng. Ltd.	27
Lawrence Bros. Millward Ltd.	104
Laystall Engineering Co. Ltd.	49
Lee, Arthur, & Sons Ltd.	66
Lewis Spring Co. Ltd., The	118
Litchfield Bros. Ltd.	109
Marbaix, Gaston E., Ltd.	Cover i
Marsden, Samuel, & Son Ltd.	116
Mavitta Drafting Machines Ltd., The	114
M.C.L. & Repetition Ltd.	116
Mek-Elek Engineering Ltd.	116
Metalastik Ltd.	45
Metropolitan-Vickers Electrical Co. Ltd.	115
Midland Motor Cylinder Co. Ltd., The	65
Minnesota Mining & Manufacturing Co. Ltd.	116
Mitchells, Ashworth, Stansfield & Co. Ltd.	109
Morgan Crucible Co. Ltd., The	93
Morleys (B'ham) Ltd.	54
Moss Gear Co. Ltd.	10
Neill, James, & Co. (Sheffield) Ltd.	19
Nestler Slide Rules	100
Newall, A. P., & Co. Ltd.	28
Newall Group Sales Ltd.	103
Newmans Tubes Ltd.	3
Osborn, Samuel, & Co. Ltd.	101
Parker Packing Co. Ltd., The	108
Park Gate Iron & Steel Co. Ltd., The	51
Patterson, W., & Co. Ltd.	100
Philidas Division of Whitehouse Industries Ltd.	17
Pioneer Oilsealing & Moulding Co. Ltd.	47
Poppy Day Appeal	102
Power Jacks Ltd.	82
Prestons Ltd.	116
Radio Heaters Ltd.	102
Ransome & Marles Bearing Co. Ltd.	69
Renold Chains Ltd.	53
Richards, Chas., & Sons Ltd.	98
Rockwell Machine Tool Co. Ltd.	98
Rotax Ltd.	57
Royal Sovereign Pencil Co. Ltd., The	106
Rubber Bonders Ltd.	73
Rye, Claude, Ltd.	116
Salisbury Transmission Ltd.	106
Salter, Geo., & Co. Ltd.	38
Selson Machine Tool Co. Ltd., The	52
Sheffield Twist Drill & Steel Co. Ltd., The	61
Shell-Mex & B.P. Ltd.	18
Silentbloc Ltd.	29
Skefko Ball Bearing Co. Ltd.	76
Small & Parkes Ltd.	110
Smith, W. H., & Son	112
Steel Co. of Wales Ltd., The	40, 41
Steel, J. M., & Co. Ltd.	84
Sterling Metals Ltd.	34
Technograph Electronic Products Ltd.	64
Terry, Herbert, & Sons Ltd.	12
Thompson, John, Motor Pressings Ltd.	80
Toledo Woodhead Springs Ltd.	14
Triplex Foundry Ltd.	86
Tucker, George, Eyelet Co. Ltd.	86
Turner Bros. Asbestos Co. Ltd.	88
Udal, J. P., Ltd.	84
Vaughan Associates Ltd.	81
Ward, Thos. W., Ltd.	78
Whittaker Hall & Co. (1929) Ltd.	89

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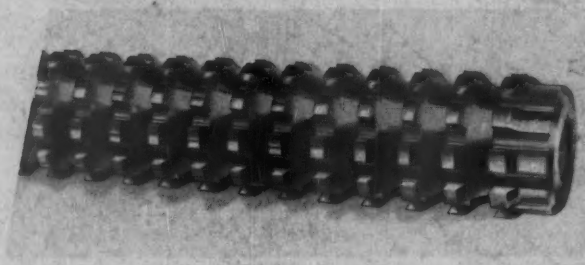

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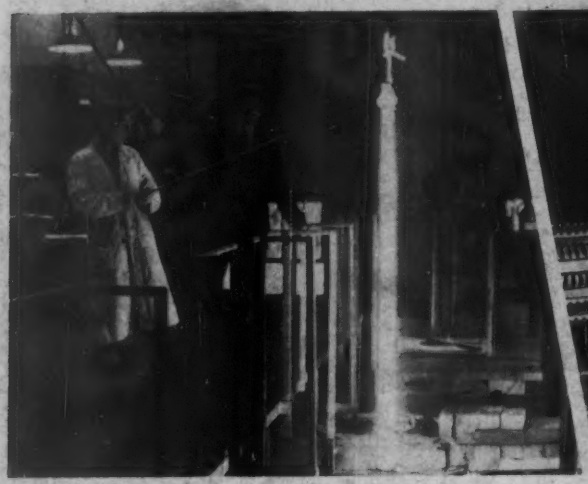
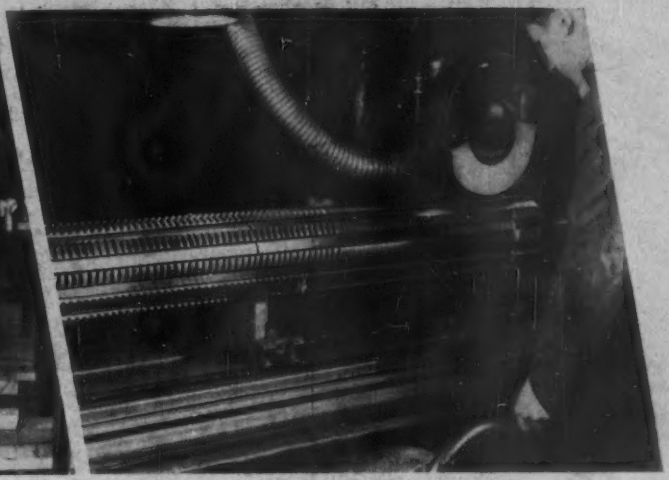
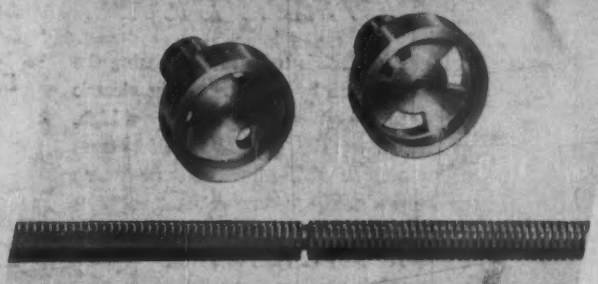




The Broach Division of B.S.A.Tools Limited is the largest plant in Europe devoted to broach production and broaching technique. It has the capacity to design and manufacture every type of broach (from those of needle size for machining metering jets to examples weighing half-a-ton or more) and is equipped for broach function-testing and for demonstrating how broaching can effect economies compared with other methods of machining. Others of its services are the regrinding of broaches and the evolving of special equipment for adapting existing machines to new broaching requirements. In fact, the B.S.A. Broach Division offers a complete service to broach users and to those contemplating the application of broaching.



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425